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Vol. 36 NEW YORK, JULY 30, 1912 No. 5

The Drink Question

HOWEVER much an engineer may strive to advance by study, or work, he is severely handicapped, if he drinks. Entirely aside from moral considerations there are powerful arguments against the immoderate use of liquor, and by this is not necessarily meant getting intoxicated. At best drinking is expensive, not only in money, but in time that might be better employed. The most that has ever been said in its favor is that it promotes good fellowship, but unfortunately that is very apt to degenerate into the acquisition of undesirable friends.

The necessity, not only of sobriety, but of total abstinence on the part of those whose duties are so important to the lives of others as those of the locomotive engineer, is universally appreciated. Only four weeks ago 46 lives were lost in a train wreck for which an intoxicated engineer is believed to have been responsible.

The most striking testimony against drink that we have read in a long time is that of John L. Sullivan, the ex-champion pugilist, in an interview in a recent issue of the *New York Sunday World*. Coming as it does from a man who has no prejudices in the matter, it is especially worth attention. Asked what was the greatest battle he ever fought he replied, "My battle with the Black Bottle." Then followed an intensely interesting story of his struggle when he quit in 1905 after 30 years of hard drinking. He acknowledged spending nearly half of all he ever earned for drink, that it prejudiced thousands of people against him, lost him hundreds of thousands of dollars in contracts, and would have been a great injury to his health but for his constant training. With it all he claims he never had the appetite, but that he drank "like most people do—for conviviality's sake, to be a good fellow."

After he quit he says: "I soon found I was feeling better. My head was clearer; my stomach better. After I had cut out the booze, I was free from even the slightest ailments." He disclaims being a temperance crank, but, as he puts it, "The man who drinks is a ——— fool." His concluding advice was: "The man, and especially the young man, who has any ambition, and hopes to succeed in life, has got to knock out the Black Bottle. The sooner he does the sooner he will succeed."

The effect of booze on the power plant of man is a decreased efficiency. Men, if the Black Bottle blocks your way, give it a knockout wallop.

Compression in Engine Cylinders

By W. J. Attick*

Under certain conditions, compression is valuable as a means of regulating the ratio of expansion and inducing quiet running.

The mathematical analysis is included to trace the action of the steam within the cylinder. Its comprehension, however, is not essential to an understanding of the conclusions.

When a perfect gas expands at constant temperature its pressure is represented by a curve whose equation is $PV = C$. In the ideal indicator diagram, Fig. 1, in which the expansion curve V_1V_2 is of the form PV = Constant, if any point D in this curve is taken, the product of the pressure and the volume is the same as the product of the pressure and the volume at any other point, as

$$PV = P_1V_1 = P_2V_2 = Constant$$

This constant varies for different temperatures and different gases.

Let the lower line, Fig. 1, represent the line of zero pressure or P_0, and the next one above, the line of atmospheric pressure or Pa. The total work done during one stroke is represented by the total area included between the admission and the expansion lines and the line of zero pressure; that is, the area $AV_1V_2\ P_0\ ECA$. That part of the diagram $ACEV_1$ under the admission line represents the work done during admission and its area is $P_1 \times V_1$. The area under the curve V_1DV_2 represents the work done during expansion. This can be found by any one of three ways: by using a planimeter; dividing the area into a number of equal parts by vertical lines, finding their mean height and multiplying this by the length of the area measured; or by means of a formula derived by calculus which states that the area under the curve is equal to $P_1V_1\ log.e\ r$, the values P_1 and V_1 being respectively the pressure and the volume at cutoff and $log.e\ r$ the hyperbolic logarithm of the ratio of expansion. The ratio of expansion r is equal to

$$\frac{P_1}{P_2} = \frac{V_2}{V_1}$$

This formula applies only to a curve of the form PV = Constant. However, in a steam engine with well fitting valves and piston, the steam in expanding follows this law so closely, due to an interchange of heat between the steam and the metal, that no great error is made in applying it. The compression curve of an indicator diagram also closely follows the same law.

In an ideal diagram for an engine without clearance or compression, the total work done during one stroke of the engine is the sum of the areas under the admission line AV_1 and the expansion curve V_1V_2, or

$$P_1V_1 + P_1V_1\ log.e\ \frac{P_1}{P_2}$$

The useful work is the total work minus that done on the steam during the return stroke. The latter is represented by the area under the back-pressure line P_2B and is equal to $P_2 \times V_2$. But

$$P_2V_2 = P_1V_1$$

hence the useful work is

$$P_1V_1 + P_1V_1\ log.e\ \frac{P_1}{P_2} - P_1V_1 =$$

$$P_1V_1\ log.e\ \frac{P_1}{P_2}$$

This is the maximum useful work obtainable from a perfect steam engine, using steam between the pressures P_1 and P_2. But steam engines are not perfect, and one of the greatest causes of loss is due to clearance. If an engine has no clearance, then for every inch of the cylinder that is filled with steam during admission the piston moves 1 in. and a proportional amount of work is done, but if there is clearance it must be filled with steam at admission pressure before any work can be done, and this steam performs work only during expansion; therefore the work it is

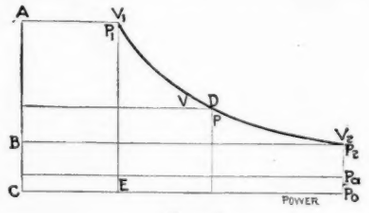


FIG. 1

capable of doing during its admission to the cylinder is lost.

It is common practice to fill the clearance with exhaust through compression and thus save whatever live steam would be necessary for this purpose. Just how much compression compensates for the loss due to clearance has not yet been decided definitely. To show the loss of useful work due to clearance, consider Fig. 2. This diagram represents the work done if there is clearance but no compression. The exhaust steam which fills the clearance space at the end of the stroke is compressed by the entering steam so that for the most part the clearance space is filled with live steam. To understand this action, suppose that when the piston is at the end of the stroke as in position D, Fig. 2, there is a thin diaphragm close against the piston and that the live steam is admitted between the piston and this diaphragm. The space Vc representing clearance, is filled with exhaust steam and as the live steam enters between the diaphragm and the piston, the former is forced back into some such position as is represented by the dotted line. The space C_1 may then be considered as filled with exhaust steam compressed to initial pressure, and the space C as filled with live steam.

To find the portion of the clearance space filled by live steam, consider the exhaust compressed along the line AB, which would be a curve following the same law as the expansion curve or PV = Constant. In Fig. 2, V_1 is the volume of steam admitted up to the point of cutoff and is the same as V_1 in Fig. 1; Vc is the clearance space; V_4 the part of the clearance space filled by live steam; V_3 that filled with compressed exhaust; V_5 is the total volume of steam in the cylinder at cutoff, and V_2 the total volume after expansion. It is assumed that expansion is carried to back pressure.

The work done during the admission period would be

$$P_1\left(V_1 - Vc + Vc\frac{P_2}{P_1}\right) =$$

$$P_1V_1 - P_1Vc + P_2Vc$$

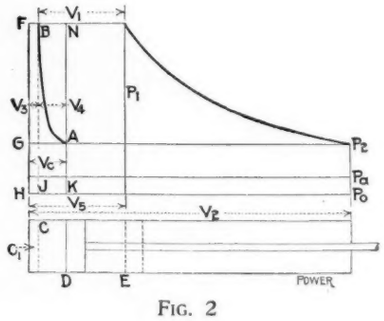


FIG. 2

The total volume of steam at cutoff would be

$$V_5 = V_1 + Vc\frac{P_2}{P_1}$$

and the work done during expansion would be from the general formula, $PV\ log.e\ r$,

$$P_1\left(V_1 + Vc\frac{P_2}{P_1}\right) log.e\ \frac{P_1}{P_2} =$$

$$P_1V_1\ log.e\ \frac{P_1}{P_2} + P_2Vc\ log.e\ \frac{P_1}{P_2}$$

The total work during one stroke is represented by

$$P_1V_1 - P_1Vc + P_2Vc + P_1V_1\ log.e\ \frac{P_1}{P_2} +$$

$$P_2Vc\ log.e\ \frac{P_1}{P_2}$$

The negative work done during exhaust would be

$$P_2\ (V_2 - Vc) = P_2V_2 - P_2Vc = P_1V_1$$

The useful work done during one stroke would be

$$P_1V_1 - P_1V_c + P_2V_c + P_1V_1 \log_e \frac{P_1}{P_2} + P_2V_c \log_e \frac{P_1}{P_2} - P_1V_1 = -P_1V_c + P_2V_c + P_1V_1 \log_e \frac{P_1}{P_2} + P_2V_c \log_e \frac{P_1}{P_2}$$

This expression may be considered as made up of two parts

$$\left(P_1V_1 \log_e \frac{P_1}{P_2}\right)$$

and

$$\left(P_2V_c + P_2V_c \log_e \frac{P_1}{P_2} - P_1V_c\right)$$

The first part is the expression for the useful work done with the volume V_1 of live steam when there is no clearance; the latter part may be negative or positive so far as the expression is concerned, depending on whether

$$P_1V_c \text{ or } \left(P_2V_c + P_2V_c \log_e \frac{P_1}{P_2}\right)$$

is the greater. If the value of this expression is negative, then the useful work obtained from a given volume V_1 of

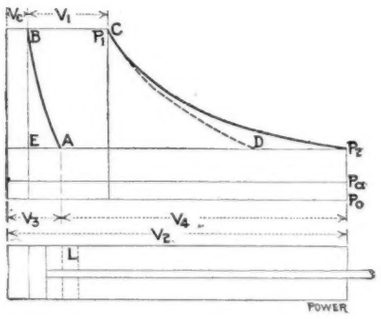


FIG. 3

steam in an engine with clearance is less than that obtained from the same amount of steam under the same conditions except with no clearance.

By referring to Fig. 2, it may be seen that

$$P_2V_c \log_e \frac{P_1}{P_2} = \text{area } ABJKA$$

so that

$$P_2V_c + P_2V_c \log_e \frac{P_1}{P_2}$$

represents the area *FHKABF* and is always less than P_1V_c by the area *ABNA*. This area represents the useful work obtained from a volume V_1 of live steam in an engine with clearance, less than that obtained under the same conditions but without clearance.

That compression under certain conditions compensates for the loss due to clearance is shown by Fig. 3, which represents the work done during one stroke with a volume V_1 of live steam but with compression to initial pressure. When the piston on the return stroke arrives at the position *L*, shown in dotted lines, the exhaust valve closes and the steam in the cylinder is compressed along the line *AB*, so that when the piston is at the end of the stroke the pressure is equal to the initial pressure. Then consider, as before, a diaphragm placed against the piston and that the live steam is admitted between the diaphragm and the piston. As the pressure in the clearance volume is equal to the initial pressure, the diaphragm will not move during admission, and the only live steam required is that necessary to fill the space back of the piston from the beginning of the stroke to the point of cutoff. The work done during admission would be P_1V_1. The total volume of steam in the cylinder at cutoff would be $V_1 + V_c$ and the work done during expansion would be

$$P_1 (V_1 + V_c) \log_e \frac{P_1}{P_2} = P_1V_1 \log_e \frac{P_1}{P_2} + P_1V_c \log_e \frac{P_1}{P_2}$$

The total work then is

$$P_1V_1 + P_1V_1 \log_e \frac{P_1}{P_2} + P_1V_c \log_e \frac{P_1}{P_2}$$

The negative work during the return stroke would be that during the exhaust period plus the work done in compressing the steam. The work during the exhaust period would be

$$P_2V_4 = P_1V_1$$

The work of compression is

$$P_2V_3 \log_e \frac{P_1}{P_2} = P_1V_c \log_e \frac{P_1}{P_2}$$

The total negative work then is

$$P_1V_1 + P_1V_c \log_e \frac{P_1}{P_2}$$

and the useful work would be

$$P_1V_1 + P_1V_1 \log_e \frac{P_1}{P_2} + P_1V_c \log_e \frac{P_1}{P_2} - P_1V_1 - P_1V_c \log_e \frac{P_1}{P_2} = P_1V_1 \log_e \frac{P_1}{P_2}$$

This is the same expression that was derived for the useful work under the same conditions and the same volume V_1 of live steam except that in the first case there was neither clearance nor compression. It is seen that, working under theoretical conditions, expanding to back pressure and compressing to initial pressure will neutralize any loss due to clearance.

This is true, however, only if the mechanical efficiency of the engine is perfect. The steam in the clearance space in expanding furnishes the power necessary to compress it. If the expansion is not complete it will not do this.

It will be noticed that in each of the three cases discussed the negative work done during the exhaust period is the same, $P_2V_2 = P_1V_1$, and this is also true only when the expansion is complete. As the useful work represented in Figs. 3 and 1 is equal, the areas AV_1V_2B (Fig. 1) and $ABCP_2$ (Fig. 3) are equal, and if the expansion curve of Fig. 1 is drawn on Fig. 2, as shown by the dotted line *CD*, then length P_2D will equal *AE*, and area *ABEA* will equal CDP_2C.

In an engine taking steam full stroke, $V_1 = V_2$ and the useful work done would be

$$P_1V_1 - P_2V_2 = (P_1 - P_2)\ V_1$$

with neither clearance nor compression. If in this case there were clearance and compression, the clearance space would be filled with compressed exhaust steam at the expense of energy taken from the flywheel, and as there is no expansion this compressed steam could do no work.

In ordinary cases, steam is not carried for full stroke nor expansion to the back pressure, although the direct-acting pump is an instance of the former, and an automatic cutoff engine running light, of the latter. As with complete expansion, compression to initial pressure completely neutralizes the loss due to clearance, and with no expansion, compression increases the loss; then for each different ratio of expansion between these extremes there must be some degree of compression which is the most economical.

A general expression for the useful work under any condition could be derived, but it would be complicated, and

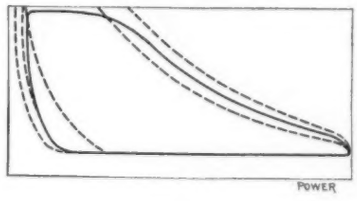


FIG. 4

it is much easier to find the proper amount of compression by drawing a theoretical diagram with various compression curves, and the corresponding expansion curves, keeping the volume V_1 up to the point of cutoff the same. Then with a planimeter measure the areas representing the useful work for the different degrees of compression, and that for which this area is the greatest will be the most economical compression to use.

If on an actual indicator diagram, Fig. 4, a series of compression curves and corresponding expansion lines are drawn, keeping the area constant, it will be found that the admission line V_1 will be shorter for one pair of curves than for any other pair. This then will be the proper compression for this particular load.

If an engine is overloaded and cutting off late, then low compression is to be desired. Cutting down the compression and leaving the ratio of expansion the same would increase the efficiency, but in the actual engine it would also give a better ratio of expansion. On the other hand, an engine may be underloaded, and with low compression will be exhausting below the back pressure, and there will

be a negative loop in the diagram which perhaps can be removed by increasing the compression. These effects of compression are entirely separate from its effect in filling the clearance volume.

Also, with the single-valve automatic engine, in which the compression changes with the cutoff, this change of compression aids the regulation as the governor does not have to move so far to change the area of the diagram a given amount as when the compression is constant. This is one reason why the single-valve engines usually regulate better than four-valve engines.

Commercial Efficiency

In what has been said so far, the thermodynamic efficiency of the engine has been considered and the results were derived by assuming theoretically perfect conditions. In actual practice it is commercial efficiency which is desired, and the conditions are not ideal. Due to heat absorption by the metal of the cylinder and piston and to mechanical losses,

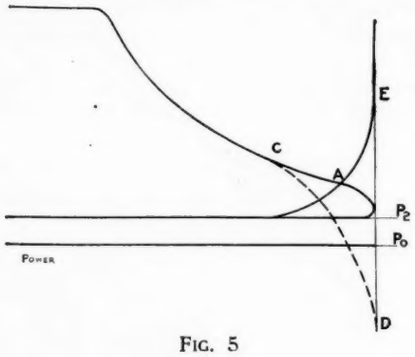


Fig. 5

not more than about 85 per cent. of the energy that is required to compress the steam can be recovered.

The claim is made that compression, by warming the cylinder, prevents to a certain extent the condensation of the live steam entering the cylinder, which is probably true, but this operation is effected by a double transformation: first, of the energy of the steam into the mechanical energy of the flywheel, and then from the flywheel to heat energy again, both transformations being made at low efficiencies. It would seem therefore that it would be better to take the heat directly from the boiler for this purpose.

So far as economy is concerned, it would appear that the greatest benefit of compression is due to its ability to regulate the ratio of expansion to a limited degree with single-valve engines. Wherever an engine is to operate under a steady load, some form of four-valve engine, with low compression, properly proportioned cylinders and tight valves and piston will give the best results, especially if it is to be run noncondensing. If, however, the load is variable, with the engine running frequently much underloaded, it is probable that the single-valve automatic engine will do better than the four-valve type.

With unbalanced or only partially balanced valves when used at high pressures, considerable power is required to move them. This is noticeable in some of the late designs of four-valve engines, with Corliss-type valves and a valve-gear which gives a very marked nonharmonic motion to the valves. This effect is so great that some engines, when they are new, cannot be run with the throttle wide open with no load as there would be danger of pulling the valve-gear to pieces. It can be remedied to a certain extent by increasing the compression, for then at the time the valves open there is less unbalanced pressure on them and they move easier.

This same effect is noticed with Corliss engines working under comparatively high pressures. They must be given more compression than at low pressures or the amount of oil required to keep the valves running smoothly will be greater.

Most engine builders also desire a certain amount of compression as an aid to quiet running, particularly in engines running at comparatively high speeds. In the case of an engine running at

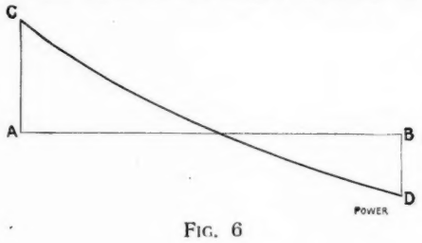


Fig. 6

high speed and with no compression, the stress at the end of the stroke must be taken on the bearings and also by the steam after admission occurs, but admission should not begin until very nearly the end of the stroke, and the reversal of pressures would be very sudden.

Effective Pressures

The ordinary indicator diagram is a combination of curves drawn during two strokes. The admission and expansion lines are drawn during one stroke and the exhaust and compression lines during the return stroke, so that the effective pressure is not measured at any instant by the vertical distance between the expansion and admission lines above and the exhaust and compression lines below, as is usually assumed. If the expansion curve for one end of the cylinder is combined with the exhaust line from the other end of the cylinder during the same stroke, then the vertical distance between these two lines would represent the true effective pressure. This makes no difference in the result if it is desired to find the power developed, but if the pressures on the pins and bearings are wanted, there is a difference. If the admission and expansion lines of one diagram are taken and combined with the back-pressure line for the other end of the cylinder, a diagram is had similar to Fig. 5.

At the point *A*, where the expansion and compression curves cross, the effective pressure on the piston is reversed; and if the cutoff is early and the compression high, the pressure will be reversed long before the end of the stroke. By laying off the effective pressure on the back-pressure line P_2 as a base, then, where compression starts, the line representing the effective pressure will leave the expansion line and extend downward, as shown by dotted line *CD*, the distance P_2D being equal to P_2E.

This gives an idea as to how the actual effective pressure on a piston varies throughout the stroke, but it does not show anything regarding the pressures on the pins and bearings. This is greatly modified by the inertia of the reciprocating parts, which may be very large. At the beginning of the stroke a certain amount of the effective pressure in the piston is used to give motion to the reciprocating parts, and while the piston

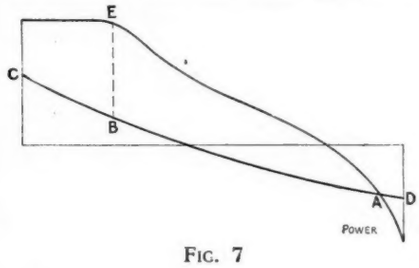


Fig. 7

is again coming to rest at the end of the stroke all this energy is given out. If the motion of the piston were the same when leaving each dead center, the inertia would affect both ends alike, but when the piston leaves the head end of the cylinder it must move faster than when leaving the crank end. This is due to the effect of the angularity of the connecting-rod. Let *AB*, in Fig. 6, represent the stroke; then the effect of inertia can be laid off vertically; if this is reduced to pressure per square inch of the piston to the same scale as that of the indicator diagram, the two diagrams showing the pressures on the crankpin due to the effects of the combined steam pressure and inertia can be combined. In Fig. 6, *AC* represents the effect of inertia at the head end and *BD* at the crank end of the stroke.

If an indicator diagram is drawn on this figure, a diagram similar to Fig. 7 will be obtained. In this diagram the vertical distances between the two lines representing the effective steam pressure and the effects of inertia would give the pressure on the crankpin at the corresponding part of the stroke. From this diagram it will be seen that the effect of

inertia is to reduce the pressure on the crankpin at the beginning of the stroke and increase it at the end.

If there were no compression, the pressure on the crank would become very large close to the end of the stroke and be suddenly reversed by the inrush of steam when the admission occurred, not only creating undue strain but causing the engine to pound if the bearings were not perfectly adjusted. The same result may be expected if the compression is very high and the load is light as the pressure is then reversed at some distance from the end of the stroke when the piston is still traveling fast.

While it can be shown that the change of pressure is comparatively gradual, the effect is the same as though the piston were traveling slowly and the pressure were suddenly reversed. This condition is seen in many single-valve automatic-cutoff engines; they will pound with light loads but will run smoothly when they have enough load to cut down the compression to the proper amount.

From the foregoing it may be inferred that the three principal advantages to be derived from the proper amount of compression are: keeping the ratio of expansion within proper limits when the load is very changeable and varies within wide limits; an aid in easily moving unbalanced valves with high pressures; and modifying the pressure on the crankpin, due to the steam pressure and the inertia of the reciprocating parts making them more uniform and preventing heavy strains and sudden reversals of pressure at the end of the stroke.

An added inference is that the benefit derived from filling the clearance space with compressed steam, so far as economy is concerned, is very small, if any, even though, as shown under theoretical conditions, it may fully compensate for the loss due to clearance.

Fan Engine Room Signals

Although the ventilating fans are of vital importance for the safe operation of a mine, the engines which drive the fans are, in many instances, unattended both night and day. Occasionally, the nearest hoisting engineer drops in to look at the engine which happens to be in his care, but at other times the engine takes care of itself.

In other instances, a fan engineer is on watch during the day-shift, but the engine is left in the care of a hoist engineer at night. There is a chance under such conditions that something may go wrong immediately following his inspection of the fan engine, and several devices have been rigged up by these engineers which will enable them to know that the engine is either running up to speed, slowed down or has stopped.

At one mine the engineer made an air pump out of a brass cylinder and fitted it with a piston and piston rod. The piston was connected to a link which was attached to a rocker-arm. This arm was pivoted at the bottom end and was made with two side arms. A carrier arm passed between them and engaged with an iron pin which ran through both arms. The carrier arm was bolted to the bottom of the engine eccentric, the motion of which actuated the air pump. The device is shown in Fig. 1.

When not in use a block is placed between the pin and the carrier arm, the block forming a riding surface for the arm.

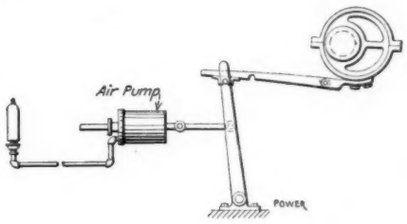


FIG. 1. AIR PUMP

When the day engineer goes off watch the carrier arm is engaged to the pin in the rocker-arm and the air pump is thus set in operation. The compressed air is carried through a small iron pipe to the hoisting-engine room where a small whistle is attached to the end of the air pipe. Thus, at every revolution of the fan engine the whistle is blown. In this case the shrill tone of the whistle had been muffled by wrapping cloth around it, so that the escaping air alone is heard by the engineer on watch, the secession of which would be instantly noted.

A second device is shown in Fig. 2. It consists of a recording gage attached to a pipe leading to the fan chamber.

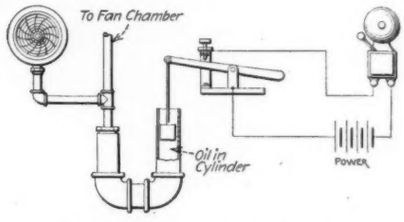


FIG. 2. DRAFT-GAGE ALARM

At the bottom of the tee connection a piece of 2-in. pipe is attached and a U-arrangement made, as shown. In the outer upright pipe a float is placed, which is connected to a contact lever by a small iron rod. The large pipe is then partly filled with oil. The contacts are connected to an electric bell terminal and batteries by wires. The outer contact block is fitted with an adjusting contact screw.

With the fan running at normal speed the pull in inches of water will lower the oil and float in the outer end of the U-connection. The adjusting screw is set so that no contact between it and the contact arm occurs while the fan is running at normal speed.

If for any reason, such as low steam pressure, the fan engine should slow down, the pull on the oil decreases and the oil rises in the outer leg. The float rises with it and moves the arm up against the contact screw, closing the circuit and ringing the bell.

A third alarm is made with a steam

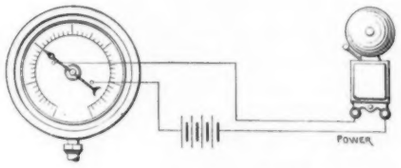


FIG. 3. STEAM-GAGE ALARM

gage connected to the engine steam pipe in the usual manner. Two spring contact pieces fit over the face of the gage, as shown in Fig. 3. If the minimum pressure necessary to operate the fan engine is 80 lb. gage per square inch, the contacts are so arranged that the gage pointer will make contact between them at that pressure. This rings a bell in the hoist-engine room and the engineer looks into the cause of the drop in steam pressure.

Hydro-electric Development near Pittsburgh

Plans are under way for constructing a hydro-electric development near Pittsburgh by the American Water Works & Guarantee Co. and its allied interests. Acquisition of water-power sites on the Cheat River, within 50 miles of Pittsburgh, has been going on for some time and it is said that with the property and rights now held, more than 100,000 hp. can be developed when the installation is complete.

The company recently offered for sale $10,000,000 in 6 per cent. cumulative participating preferred stock and it is understood that part of the proceeds will be used for the hydro-electric enterprise.

Most of the power generated can be supplied to the railway, light and power systems of the West Penn Traction Co. alone, in addition to the local demand in the West Virginia and Pennsylvania markets. A majority of the West Penn company stock is owned by the American company.

The traction company has 209 miles of track and supplies light and power in 110 cities and towns in the Pittsburgh district within a radius of 30 to 60 miles of the city. The main power plant, in Connellsville, has a capacity of 42,000 hp., which will soon be increased to 55,000 hp. The demand from the West Penn system alone will probably exceed the capacity of the first of the hydro-electric company's development.

Large Steam Turbine Development

A discussion of the present development of large steam-turbine construction and of the field of the steam turbine both in this country and in Europe.

Steam turbines are now being used for driving alternating-current generators, turbo-compressors, turbo-blowers, pumps and marine propellers, and through gearing, to furnish power to direct-current generators, rolling mills and the propeller shafts of steamships. Reciprocating engines were formerly used for such purposes, but recently this class of engine has seldom been installed except for rolling-mill work, non-condensing service as on heating systems, rope and belt drives, hoists, and in certain combinations with low-pressure turbines in marine work. The high economy of the piston-pumping engines and also of some types of air compressors, has continued their popularity in spite of the increasing competition of steam-turbine units. The steam turbine has found favor principally on account of its low cost of installation, its small floor-space requirements, its continued good steam economy over a period of years and its small operating and repair charges.

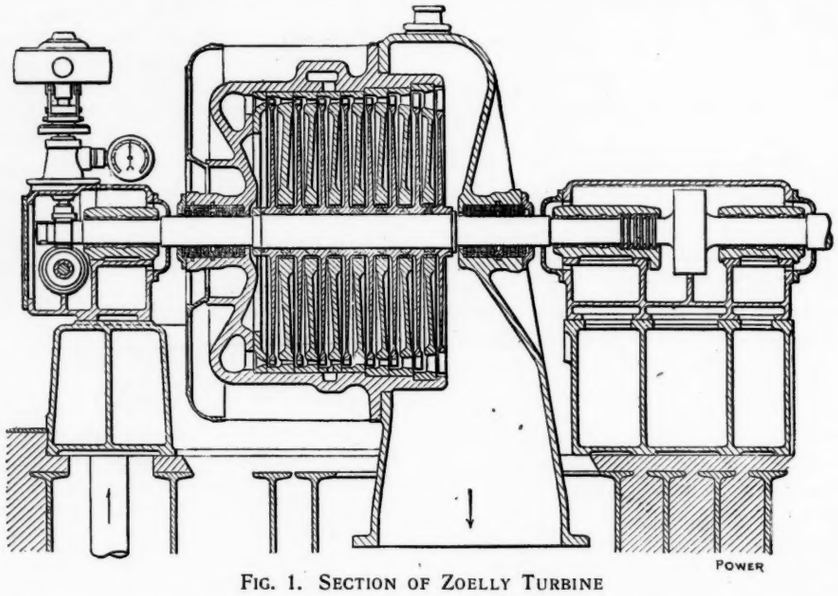


FIG. 1. SECTION OF ZOELLY TURBINE

The increased use of steam turbines in sizes up to 1000 hp. seems to have received at least a temporay check in Europe by the introduction of the new Stumpf direct-flow engine.

TYPES OF STEAM TURBINES

For the purpose of this paper, large commercial steam turbines will be divided into two classes: (*a*) fundamental types, and (*b*) modified or combined types.

The fundamental types of turbines are as follows:

a The Parsons type, which works on the so called "reaction" principle. In this type the heat energy of the steam is changed into kinetic energy, both in the stationary guide blades and in the moving blades. In other words, both sets of blades act as orifices expanding the steam through a small pressue drop. As nozzles and orifices usually have very high efficiencies, this turbine should, theoretically, prove the most economical of all types. The Parsons turbine consists of a drum, or a number of drums, carrying the blade rows which alternate with rows in the casing. The drums carry balance pistons to equalize the end thrust.

b. The Curtis type, which works on the impulse principle with high steam velocities and few stages. Each stage, however, has two or more rows of revolving blades known as "velocity rows," with intermediate rows of guide blades. The steam velocity at the beginning of each stage is high. The revolving blades are carried on disks separated by diaphragms, which extend to the shaft and carry the orifices between stages. Curtis turbines are now usually built with horizontal shafts. In American practice some sizes over 7000 kw. still have vertical shafts.

c. The Rateau turbine, which consists of a number of simple impulse wheels in series on the same shaft and separated from one another by diaphragms carrying nozzles. It operates with lower steam velocities than the Curtis and consequently has many more stages. Each revolving element caries only one row of blades.

The type of turbine known as the Zoelly belongs to the same classification as the Rateau, from which it differs only in the use of higher steam velocities, in the number of stages and in certain constructional details. Fig. 1 shows a section of a Zoelly turbine. Though this type has been widely adopted abroad, it has not yet been placed on the American market.

Each of these fundamental types is based on sound theoretical principles. In the process of manufacture and in operation, certain features have not proved entirely satisfactory, hence far-reaching modifications have been made in the design of some types of turbines. Some manufacturers have combined the characteristics of two or more types to overcome the inherent limitations of each fundamental type. A discussion of the unsatisfactory operating conditions of each type will show the reasons which led up to the changes in recent turbines and will also aid in distinguishing the novel features of new designs.

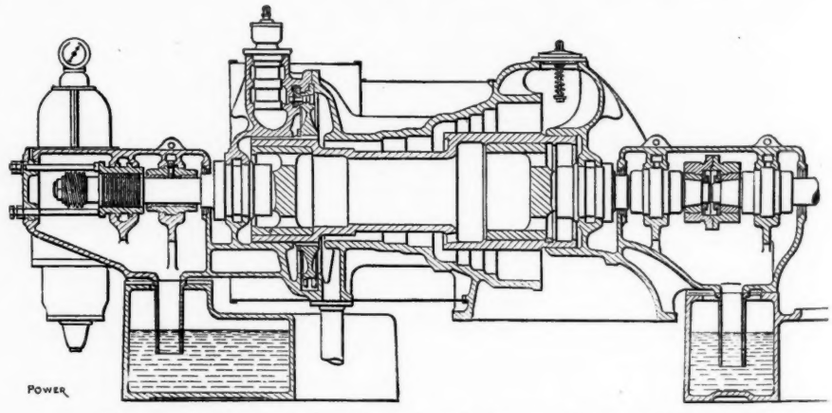


FIG. 2. SECTION OF BROWN BOVERI'S CURTIS-PARSONS TURBINE

The first row of spindle blades in a

*Abstract of paper presented by Prof. A. C. Christie at the spring meeting of the American Society of Mechanical Engineers.

standard Parsons turbine is placed on a drum of small diameter to make the blades as long as possible and to minimize the proportional leakage losses past the ends of the blades. A large number of rows is provided to keep the steam velocities low, as the blade velocities must be low with the small drum. The drop in pressure at each row is small and hence the leakage is correspondingly reduced. This construction results in a

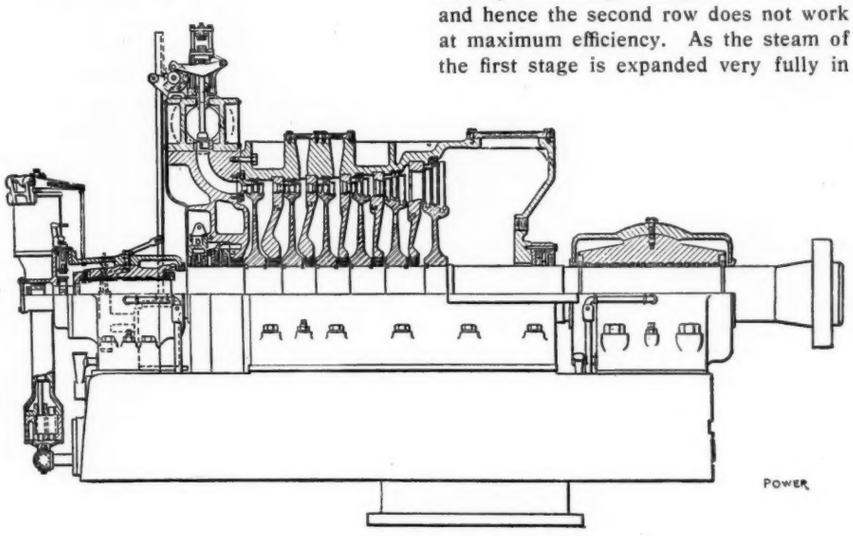


FIG. 3. HORIZONTAL SIX-STAGE CURTIS STEAM TURBINE

turbine with a long spindle and with great distance between bearings. High-pressure steam, frequently at high temperature, is admitted to the casing. Distortion of casing and spindle is thus easily conceivable in such construction, and to allow for this contingency the clearance on the ends of all blades is usually increased. This distortion may be due either to unequal heating or to the growth of the cast-iron casings. The fluid friction losses are large in this high-pressure section, for a large number of rows of blades must be revolved in steam of high density. The leakage losses and fluid friction losses in the high-pressure section, with the troubles due to distortion in the long shaft, have forced designers to modify this part of the turbine, either by new blading arangements, by dividing the total expansion in large sizes between two cylinders, or by introducing impulse blading.

The low-pressure sections of Parsons turbines have always shown high efficiencies. As low steam velocities are characteristic of this type, there is no cutting away of blade material, even with very wet steam, unless the feed water has injurious properties. This low-pressure section has therefore been altered only in details.

The presence of a large low-pressure balance piston close to the steam inlet has frequently caused serious distortions. Many builders now place this balance piston in the exhaust end, a construction due to Fullager.

The Curtis turbine utilizes high steam velocities in all stages. As steam becomes wet through expansion in the low-pressure stages, it frequently cuts the blade material considerably, although, contrary to first expectations, cutting in the high-pressure blades due to the high initial velocities is rare. The first row of velocity blades in a stage usually does the greater part of the work, and hence the second row does not work at maximum efficiency. As the steam of the first stage is expanded very fully in the nozzles, there is no high pressure or superheat in the turbine casing, or at the glands. The vertical type of unit is sometimes subject to electrical unbalancing and to other troubles peculiar to this

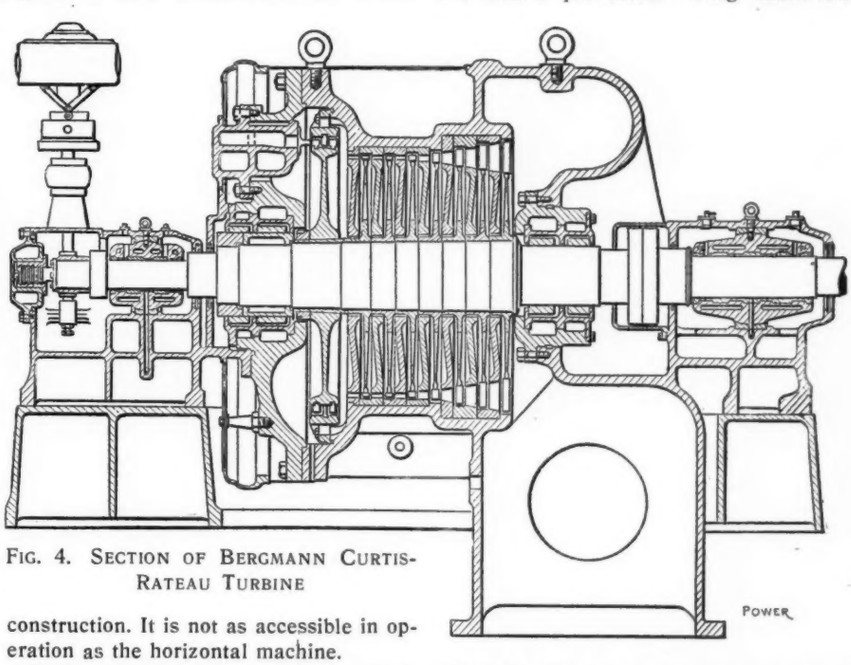


FIG. 4. SECTION OF BERGMANN CURTIS-RATEAU TURBINE

construction. It is not as accessible in operation as the horizontal machine.

Recent designs have provided for horizontal units and for the replacement of the low-pressure section by sections of other types.

The Rateau turbine has high pressures on the gland at one end. A large number of disks revolve in dense steam at the high-pressure end. It has lower steam velocities throughout than the Curtis, and consequently has no blade cutting in the low-pressure section. The clearances around the blades are large, but the shaft clearances of the diaphragms must be kept small. Some builders of this and the Curtis type have brought out new designs which employ the high-pressure section of the Curtis with the low-pressure section of the Rateau. These represent a compromise between efficiency and manufacturing costs.

Under the classification of modified or combined types, are turbines with modified Parsons, Curtis and Rateau construction, and turbines with combinations of Curtis-Parsons, Rateau-Parsons, Curtis-Rateau and Curtis-Zoelly construction, with a few special combinations. Some typical turbines of these classes are described in the following paragraphs:

The Allis-Chalmers Co. manufactures a modified Parsons turbine. High peripheral speeds are employed with a decreased number of rows. Part of the theoretical efficiency in the high-pressure end has been sacrificed by the use of fewer blade rows. Also a smaller proportion of work is done in this section than is usual in Parsons turbines. The blades all have a channel-shaped shroud. European experimenters have pointed out that better efficiencies are obtained with shrouded than with plain blades, as the so called spilling-over at the ends is prevented. Wing blades are fitted in the last low-pressure rows to care for the large volumes of low-pressure steam. The spindle is much shorter and stiffer than the standard Parsons, and hence smaller clearances can be

provided. The Fullager low-pressure balance piston is also used. The outstanding features of this design are, reaction principle with drum construction, few rows of blades with high steam velocities, short spindle construction, and employment of wing blades for high vacuum. The steam consumptions obtained show improved efficiency over standard construction.

A number of Parsons turbines have been built in which the total expansion is divided between two cylinders. One of the best known units of this type is

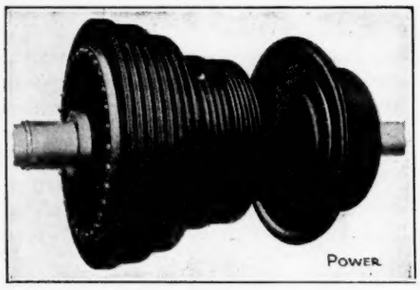


FIG. 5. SPINDLE OF CURTIS DRUM IMPULSE TURBINE

installed at the Dunstan Power Station, England. This was built by Richardsons, Westgarth & Co. on Brown-Boveri designs. This unit, however, employs a modified double-flow Parsons construction in the low-pressure cylinder.

C. A. Parsons & Co. have used cast-steel cylinders for the high-pressure part of their turbines to overcome the troubles due to deformation and growth of metal under high pressures and superheat experienced with cast-iron casings.

The Westinghouse Machine Co. has developed a double-flow machine which employs a Curtis high-pressure stage with Parsons intermediate and double-flow low-pressure sections. Its only balance piston is next to the Curtis ring. These turbines run at high speeds, have short shafts and small clearances. The steam is expanded in the nozzles, and hence there are no high pressures or temperatures in the casing. By providing two low-pressure sections, high vacuum can be economically utilized.

European builders of Parsons turbines, among whom are Brown, Boveri & Co., C. A. Parsons & Co., Franco Tosi, Sulzers, Willans & Robinson and Erste Brünner, have replaced the high-pressure sections of their Parsons turbines by a Curtis wheel with two or more velocity rows, but have retained the single-flow Parsons drum construction for the remaining portions. One of these units is shown in Fig. 2. The temperatures and pressures in the casing are low as the steam is expanded in the nozzles. The distance between bearings is decreased, the shaft is stiffer and clearances are smaller than in the standard Parsons turbine. The Fullager balance piston is used in the turbines of several of these builders. Turbines thus built have shown some exceptionally good efficiencies.

Compared with the Westinghouse machine, the leakage at the end of the low-pressure blading is less than in the double-flow section. On the other hand, the Westinghouse machine has smaller balance-piston losses and can also utilize the highest vacuum at better efficiency.

Melms & Pfenniger employ a drum impulse section of about five stages instead of a Curtis ring on some of their large turbines. They claim improved efficiency from this construction, though high-pressure and high-temperature steam are introduced into the casing.

The General Electric Co. now manufactures a horizontal Curtis turbine in all sizes up to 7000 kw., a section of which is shown in Fig. 3. This embodies all the essential features of the Curtis design. Compared with the vertical type, it provides easier access to all working parts, such as governor, bearings, valves, etc., and allows a better survey of the unit. The machine can also be dismantled and its internal parts examined with less trouble. The oiling problem is very simple compared with the vertical units, as there is no step bearing. It is possible that only horizontal units of all sizes will be built in the near future. The Algemeine Elektricitäts Gesellschaft also builds similar turbines up to 1000 kw. capacity.

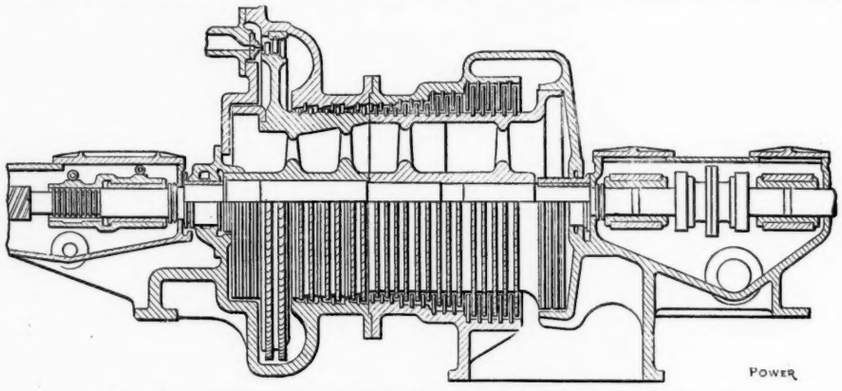


FIG. 6. SECTION OF CURTIS DRUM IMPULSE PARSONS TURBINE

Many manufacturers in Europe are now building a turbine of the type shown in Fig. 4. This consists of a Curtis high-pressure with Rateau or Zoelly low-pressure sections, the number of stages depending on the size of the machine, the speed and the steam conditions. The pressure and temperature in the casing are low, as a consequence of the expansion of the steam in the nozzles. The steam velocity is moderate in the low-pressure section. This machine is longer than the simple Curtis but shorter than the Rateau. Its builders claim improved efficiencies over either of the fundamental types.

Belliss & Morcom, of Birmingham, England, have introduced a turbine, the spindle of which is shown in Fig. 5. It consists of a high-pressure Curtis stage with a low-pressure drum construction like Parsons, but fitted with impulse blading on the spindle and blades forming expansion parallel wall nozzles in the casing. All other impulse turbines employ diaphragms extending to the shaft. A Belliss turbine has more rows than a corresponding Rateau turbine, and hence less pressure drop through each set of guide blades. The net leakage loss may thus be lower than the Rateau, even though the leakage area is greater. This turbine is essentially a Curtis-Parsons construction, with few rows of blades and with the expansion taking place in the stationary blades only.

A turbine, shown in Fig. 6, has recently been patented in England, which has some features of unusual interest. The high-pressure stage is Curtis, the intermediate section is of a drum-impulse type with three stages, while the low-pressure portion is Parsons, which may be also double-flow. This combines features of all three fundamental types, and it will be interesting to see what results are obtained in practice by such an arrangement.

(The paper continues with a discussion of turbine details, such as blading, nozzles, bearings and lubrication, spindle construction, packing glands, thrust bearings, governing devices, casings, bedplate and turbo-generators, and contains a voluminous table in which are given the characteristics of the turbines of different builders as well as the table of test results reproduced in POWER, June 11, page 832. It touches also upon low- and mixed-pressure turbines, turbo-compressors, turbine-driven pumps, geared turbines, and marine turbines.)

TREND OF TURBINE DEVELOPMENT

The cost of manufacture is very important in determining the future development of the steam turbine. Types such as the original Parsons and the Rateau, while inherently very efficient, cost too much to build to compete with the newer combined types.

The writer offers as his opinion that the combined types, such as the Curtis-Parsons, the Curtis-Rateau and also the

Curtis-Rateau-Parsons, will soon supersede the simple types. Probably the Curtis turbine will eventually be built only in horizontal units and will gradually be modified to a Rateau or even a drum-impulse construction in the low-pressure sections. The freedom from close adjustment in impulse turbines and the recent improvements in blading materials will greatly increase the use of this type, although Curtis-Parsons turbines are said to be cheaper to manufacture. In actual operation, it is an open question among engineers whether the reaction turbine has a higher commercial efficiency than the impulse type, and hence buyers usually consider first cost and personal preference only.

Turbines will probably be made shorter with very stiff shafts. With this construction many of the earlier blading troubles will disappear. But the peripheral speeds will also be increased and this will involve the development of suitable blading material and ways of holding blades that will satisfy these new requirements. Recent results seem to indicate that improved efficiency may be looked for with increased blade speeds.

Several impulse turbines have been built recently in Europe where the expansion was not complete in the nozzle, so that part of it took place in the first moving blades. Some large Curtis turbines recently installed in America are said to have Parsons blading in the last stage. These developments would indicate a movement to introduce reaction principles in impulse turbines, and further illustrate the tendency to merge types.

The hope of further improvement in efficiency lies in extensive study, particularly of the action of the steam during its passage through the moving and stationary blades, of the effect of form of blades, passages and casings, and of various forms of baffles and balance pistons to prevent leakage. Such research work has not been carried out up to the present time by most manufacturers, largely on account of the extreme care and heavy expense involved in such tests. The present state of development has been largely one of cut-and-try. The increasing competition of the gas engine and the possible development of a satisfactory gas turbine will force manufacturers to develop their turbines to the greatest degree of economy.

With regard to detail, simplicity will be the leading consideration. With the introduction of Curtis high-pressure stages, nozzle governing will undoubtedly be used to an increasing extent, though the results obtained by Westinghouse, Zoelly and Bergmann with simple throttling governors, raise a question as to whether the additional complication of nozzle governing will pay. Oil relays will probably replace all other systems of governing on account of their simplicity and reliability. The simple and efficient centrifugal oil-pump governor of Sulzer appears to be an improvement of considerable moment, and will probably be extensively used.

With the development of suitable gearing for steam turbines, their field of application has been greatly increased and turbines will shortly be used where engineers today would consider them utterly unfit. Low-pressure turbines will continue to be installed in plants where reciprocating engines are still in operation and also where large quantities of waste heat are available. The low- and mixed-pressure types of turbines will find extended use in connection with heating systems, evaporators, etc.

The development of the past ten years has been truly marvellous. No great gain in thermal efficiency seems possible, so that future improvements will be largely along the line of detail construction and modification.

Explosion of Air Receiver Due to Oil

By Frank Richards

At a temperature of about 500 deg. the oil which coated the inside of the head became ignited and heated the latter to a point at which it dished outward, the sudden shock shearing off the rivets.

A small air receiver connected with a portable gasoline-engine-driven air compressor employed on street gas-pipe work exploded on June 24, near the entrance to Greenwood Cemetery, Brooklyn. The occurrence exhibited a number of interesting features illustrative of the conditions under which such accidents may occur.

There seems to have been no sudden and enormous increase of pressure, such as would be caused by the ignition of an explosive mixture composed of volatilized constituents of the oil combined in suitable proportion with the compressed air; the normal working pressure which the receiver usually carried was probably not much if at all exceeded; nevertheless it seems to have been ultimately a simple pressure explosion, but accompanied by abnormal conditions which provided the opportunity.

The receiver was suspended horizontally under the frame which carries the gasoline engine and the compressor. It was 6 ft. long and 2 ft. in diameter, with dished heads; the convex side of the head was outward at one end and the concave side outward at the other end, which is the too common practice, just for convenience in riveting in the last head, and for no other good reason. The air entered the receiver from the compressor about 2 ft. from one end and half way up, and was taken out at the other side just opposite the inlet (see Fig. 1). A pop safety valve on the pipe from the compressor was set at 110 lb., and the usual working pressure was about 100 lb.

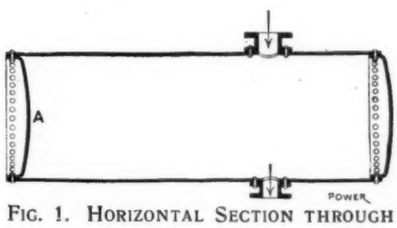


FIG. 1. HORIZONTAL SECTION THROUGH RECEIVER

As disclosed after the explosion, the entire interior surface of the receiver was covered apparently with all the oil it could carry. If the operator had been told to use all the oil he could, instead of as little as possible, the surface could not have been more oily. From time to time all the oil and water which might have collected at the bottom of the receiver should have been drained off; this may or may not have been done, but evidently it would not have made the surfaces clean, nor have freed them from the gummed or thickened oil with which they were coated.

The inner surface, *A*, Fig. 1, of the head which blew out, however, was dry and absolutely clean and its color and condition showed that it had been red hot. Evidently a local fire had raged, perhaps only for a minute or so, on this (then oil-coated) surface, burning the oil off and suddenly heating the plate; the fire did not have time to spread to the adjacent surface of the shell, when the explosion occurred.

In the explosion several curious things happened which it is worth while to note carefully. The head became so hot that, as it was under about all the air pressure it could stand, the dishing of the head was suddenly reversed, and the originally convex surface became concave, as shown in Fig. 2, this being a view of the inner surface. That this is the inner surface of the head is corroborated by the lip at the edge of the sheet. The ridge which is dented across the face was evidently

caused by something against which the head struck as it flew off.

It is easy to imagine that this reversal of the dishing of the head took place with a considerable snap, the shock and momentum, in addition to the aggregated air pressure of 45,000 lb., being sufficient to cause the head to let go. When it did, not a single rivet was sheared. It will be noticed from Fig. 2 that when the reversal of the dishing occurred, the rivet-carrying lip of the head was bent outward all the way around, so that the opposite sides, instead of being parallel, stood off 30 to 45 deg. from the normal position, and the end of the sheet to which they were attached (see Fig. 3) bent outward in the same way. There were eighty ⅜-in. rivets, and when the lip was bent out their heads were pulled off, dropping outside the shell, and then the rivets pulled out of the holes. The fracture of the rivets was square across under the heads and flush with the outside of the shell. Of the 80 rivets, three retained their heads and those tore notches out of the shell.

When the head flew off, the body of the receiver was driven in the opposite direction, but could go only a few inches when it struck the axle of the machine, which dented the head as shown in Fig. 4. Other than this the shell seems to have received no damage.

Undoubtedly receivers have often taken fire internally, as indicated in this case, but having a sufficient factor of safety, in spite of the heating of the metal, they have been able to withstand the stress and no accident has resulted.

The interesting question is as to the conditions causing, or accompanying, these ignitions of the oily surfaces, and usually more or less mystery is made of this phenomenon when it occurs. Compare the oil tempering of steel springs. The spring is first heated in the fire to a bright red and then plunged into oil and cooled. This leaves it hard and brittle, and it is then drawn to the proper temper by "blazing off." To do this, the spring, while dripping with oil, is held over the fire, which must be without flame, and is slowly and carefully heated until the oil on the spring bursts into flame. The oil is not ignited by the fire, the flame merely appearing when the spring reaches the right temperature. The oil will keep burning until consumed, or the spring, while blazing, may be dipped into the oil and the flame will go out, but if quickly withdrawn the flame will appear again. At a certain temperature oil will ignite spontaneously, and this temperature is so fixed, differing, however, for different oils, that it has been taken for generations as an index of the heat required for the tempering of springs, swords, etc.

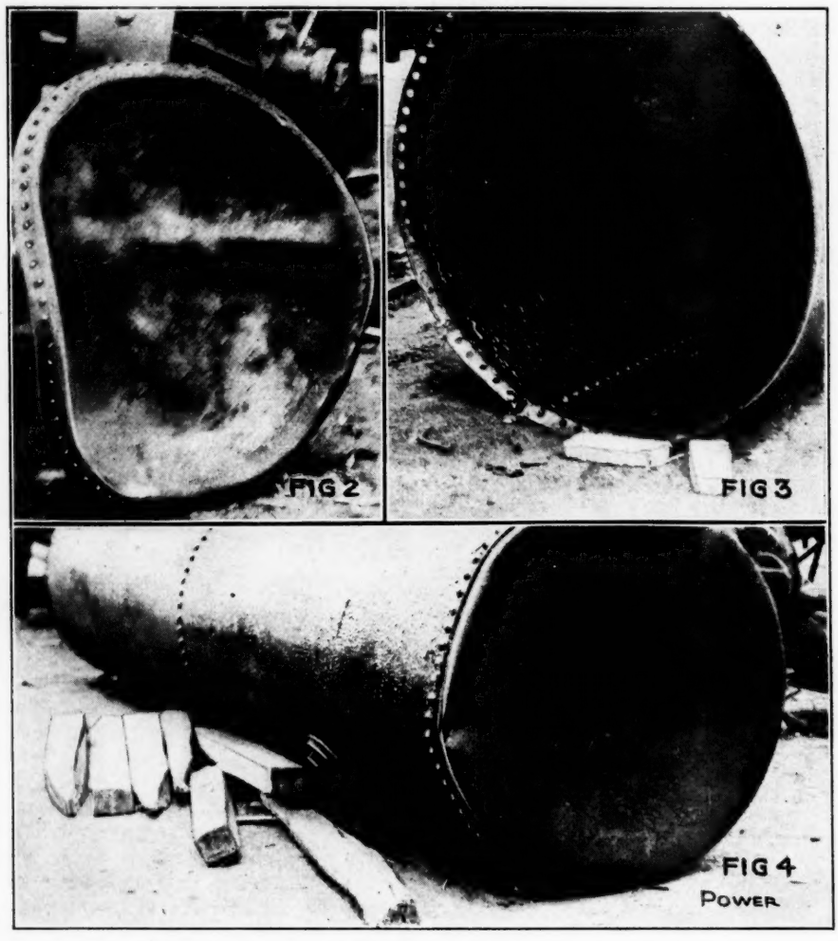


FIG. 2. INSIDE FACE OF HEAD AFTER EXPLOSION
FIG. 3. SHOWING FLARING END OF SHELL WITH RIVETS PULLED OUT
FIG. 4. OTHER END OF RECEIVER

Precisely the same thing that happens in the oil-tempering of springs will happen to the oil-coated surfaces of air reservoirs, pipes, etc., when they get hot enough. The oil will take fire of itself without any spark, flame or other extraneous source of ignition. The oil-covered steel spring, when the oil upon it takes fire in the open air, is below a visible red heat, and that begins at about 700 deg. F. In the air receiver under consideration, the temperature due to compression may have been as high as 500 deg. With air at 60 deg. when compression begins, the theoretical terminal temperature, after adiabatic compression to 105 lb., is 496 deg. This compressor was running in bright sunshine on a hot day, and the intake air passed close to the already heated air receiver before entering the cylinder, so that its temperature at the beginning of compression was presumably above 100 deg., and its terminal temperature above 500 deg. There being nothing at the end of the receiver to cause any active movement of the air in contact with the oily surface, may have offered special opportunity for the atoms of carbon and oxygen to unite. Furthermore, since the air was compressed to 8 atmospheres the oil was exposed to 8 times the amount of oxygen that it would have at atmospheric pressure. This may partly account for the spontaneous ignition of the oil at a lower temperature and its burning more fiercely after ignition.

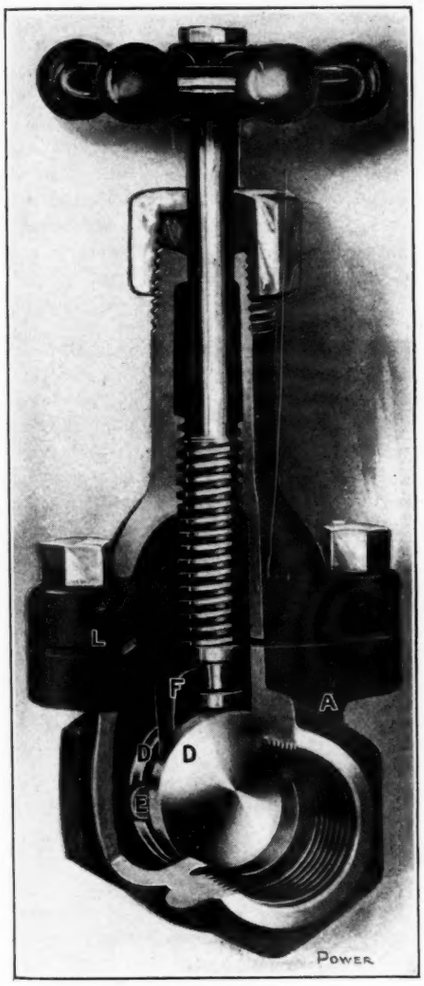
An account of an ignition of this character recently appeared in *Mines and Minerals*, and was reprinted in *Compressed Air Magazine*, June, 1912. No damaging explosion occurred, however. It was in connection with a four-stage compressor employed for charging locomotives. The ignition took place after the fourth stage of the compression, when the air was at a pressure of 65 atmospheres, and a recording thermometer, which was attached and in operation, showed that the ignition took place at 270 deg., which seems to corroborate the assumption that the higher the pressure the lower will be the temperature at which spontaneous ignition will occur.

This discussion distinctly avoids any consideration of the more disastrous class of compressed-air explosions caused by the ignition of an explosive mixture of air with some of the volatilized constituents of the oil, as there seems to have been nothing of that in this case.

"Cino" Iron Body Gate Valve

The new "Cino" iron-body gate valve is shown in section herewith. The bronze disks *D* have a ball-and-socket bearing at their center, suspended from and held in position by a malleable iron, wedge-shaped holder with lugs *E*, on either side which travel in corresponding grooves in the body.

In closing the valve, the disks automatically rock into position and adjust themselves to the taper of the seats. The



CINO GATE VALVE

spreading action of the ball-and-socket bearing is supplemented by the wedging action of the disk holder *F*, which affords any further tightening necessary after the disks have seated.

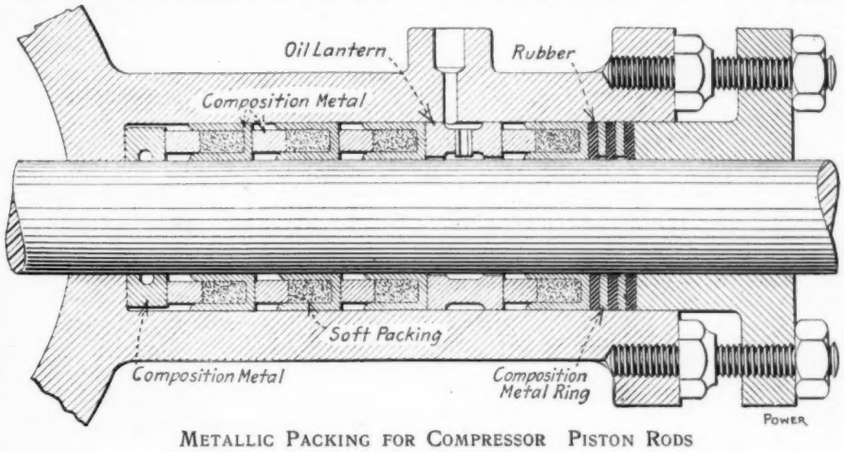
The flexibility of this double-disk mechanism compensates for any distortion or derangement of the angle of the seats, even though one disk may be temporarily obstructed by particles of dirt or scale. The valve takes pressure from either side and has a full and unobstructed opening.

Should anything accumulate in the pocket formed by the seats, it is crowded upward between the two disks when closed, and on the next opening is carried up and washed away. The two disks are coupled to the disk holder so that they are free to revolve, opening or closing and, actuated by the pressure, make almost a complete revolution, which automatically grinds in the disks and cleans the seating surfaces of dirt or scale.

It is not always necessary to replace worn disks with new ones, as they can be reground. Both disks are interchangeable. The valve is manufactured by the D. T. Williams Valve Co., Cincinnati, Ohio.

Goetze Compound Metallic Packing

This packing is constructed for compressor work and is made of composition metallic and rubber and soft packing rings. The rings are arranged as shown in the illustration. The composition metal rings are made U-shape, in the hollow



METALLIC PACKING FOR COMPRESSOR PISTON RODS

of which soft packing is placed. The packing is guaranteed for two years and is manufactured by the Goetze Gasket & Packing Co., New Brunswick, N. J.

Albany Sectional Dumping Grate

A new dumping grate recently gotten out by George H. Thacher & Co., Albany, N. Y., is illustrated herewith.

The grate frame, designed to come below the fire surface, is made with lugs which fit in grooves in the rocker bar where they are fastened in place. Each frame contains several sections of grates.

The side frame is made with lugs on the inner side against which projections on the rocker bars rest when the grates are locked in a level position. The grates assume an angle of 65 deg. when dumping, but can be adjusted to a dumping angle of 90 deg.

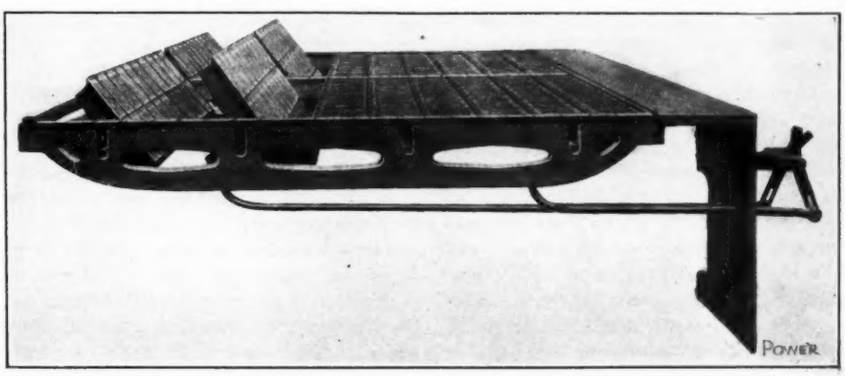
All parts are rugged in construction and the sectional fire surface is designed to prevent warping of the grate bars or side frames. Each grate overlaps the next one at the rear and the front grate overlaps the rear sectional end piece.

The Baume Hydrometer

BY LOUIS GROSSBAUM

A hydrometer is an instrument for determining the specific gravity of liquids, and is based on the principle that the weight of a floating object must always equal the weight of the liquid it displaces. The commonest is the Baumé hydrometer, of which there are two kinds, one for liquids heavier than water and one for liquids lighter than water.

The instrument consists simply of a glass float with a vertical cylindrical stem, graduated in units known as degrees. The float is immersed in the liquid, and the stem kept vertical by being weighted with a little mercury in the lower bulb. The depth to which the stem



ALBANY SECTIONAL DUMPING GRATE

sinks indicates the density of the liquid in degrees Baumé.

To convert degrees Baumé into specific gravity, the following formulas are used:

For liquids heavier than water,

$$Sp.gr. = \frac{145}{145 - deg.\ Baume}$$

For liquids lighter than water,

$$Sp.gr. = \frac{140}{130 + deg.\ Baume}$$

For instance, Fullerton oil, with a density of 19 deg. Baumé has a specific gravity of

$$\frac{140}{130 + 19} = 0.9396 \text{ or } 0.94$$

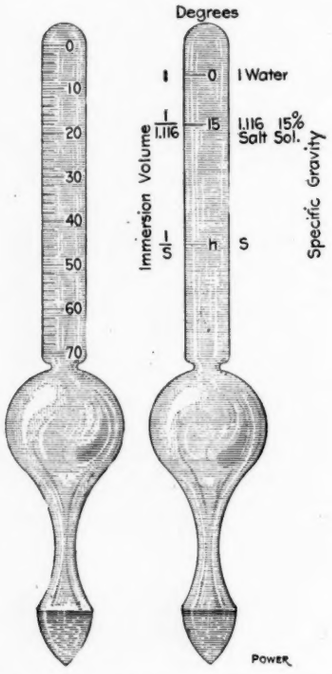


FIG. 1. HEAVY-LIQUIDS HYDROMETER

To derive these formulas the method of calibration of the instruments must be understood. Consider first the "heavy-liquid" hydrometer, Fig. 1. This instrument is so constructed as to sink to a point near the top of the stem when immersed in distilled water at 12 deg. C. (53.6 deg. F.) This is the zero point of the scale. When immersed in a solution of 15 parts of common salt to 85 parts of water (sp. gr. 1.116) the instrument floats with more of the stem exposed than before, because less displaced liquid will equal the weight of the hydrometer. The point on the stem now at the level of the liquid, by convention is designated 15 deg. The space between this point and zero is divided into 15 equal parts, and the graduations are continued to the bottom of the stem.

To find the specific gravity corresponding to any degree on the scale, consider as unity the volume displaced when the instrument is immersed in water. This volume varies inversely with the specific gravity of the liquid, as is readily seen from the relation:

$$Sp.gr. \times volume = constant \times weight = constant$$

or

$$Volume = \frac{constant}{sp.gr.}$$

Consequently, when the instrument is placed in a liquid of specific gravity S, the volume immersed will be proportional to $\frac{1}{S}$, and the difference between this and the original volume $1 - \frac{1}{S}$. Assume that the hydrometer sinks in this liquid to a point h degrees below the original zero mark. Since the stem is cylindrical, this height h evidently must be proportional to the volume difference $1 - \frac{1}{S}$.

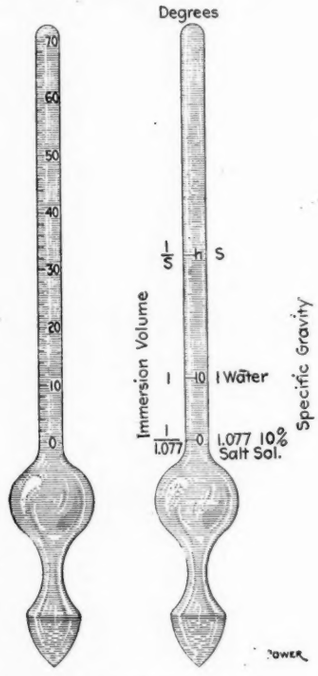


FIG. 2. LIGHT-LIQUIDS HYDROMETER

From the calibration it is known that when

$$h = 15\ deg.\ 1 - \frac{1}{S} = 1 - \frac{1}{1.116}$$

The general proportion therefore can be deduced

$$\frac{h}{1 - \frac{1}{S}} = \frac{15}{1 - \frac{1}{1.116}}$$

This proportionality may be readily seen from the diagram. The equation reduces to

$$S = \frac{145}{145 - h}$$

very nearly, where h is degrees Baumé.

The "light-liquids" hydrometer, Fig. 2, is similar, except that it is designed to sink to a point near the bottom of the stem when immersed in a solution of 90 parts of salt to 10 parts of water (sp.gr. 1.077), and this is taken as the zero point. The second calibration point is obtained by immersing in distilled water. The stem sinks deeper than before, and the point reached is designated as 10 deg. The graduations are continued to the top of the stem, the numbering being upward instead of downward as in the heavy-liquid instrument.

If h is measured from the zero point, the relation exists that h is proportional to the change in volume, which is now

$$\frac{1}{S} - \frac{1}{1.077}$$

For water,

$$h = 10 \text{ and } S = 1$$

so that

$$\frac{1}{S} - \frac{1}{1.077} = 0.0715$$

and the proportion obtains that

$$\frac{h}{\frac{1}{S} - \frac{1}{1.077}} = \frac{10}{0.0715}$$

From which

$$S = \frac{140}{130 + h}$$

very nearly, where h is degrees Baumé.

The above formulas are those generally used in America. Others have been worked out from different points of calibration, and using different (and usually more correct) values of the specific gravity for the calibration points. This accounts for discrepancies in different tables for specific gravities, corresponding to degrees Baumé. It should also be noticed that the formulas are used to apply to liquids at 60 deg. F. For other temperatures corrections must be made, as explained in POWER, Mar. 19, 1912.

New Generators for Keokuk

At the Schenectady works of the General Electric Co., the 7500-kw. generators for the Keokuk, Iowa, hydro-electric development are nearing completion.

Because of the slow speed of the low-head turbines, these generators are of great size for their rated capacity. They are of vertical-shaft type, 32 ft. in diameter and 12 ft. high, weighing over 600,000 lb. assembled. The turbines are of single-runner type with a 25-in. shaft revolving at 57.7 r.p.m. In spite of a low rotative speed, the peripheral velocity rises to about 4620 ft. per minute or 52.5 miles per hour. The turbine runners are about 15 ft. 8 in. in diameter and weigh 140,000 lb. each. The complete runner, including the generator rotor, weighs about 550,000 lb.

Canadian patent No. 138,432, says a contemporary, covers "an explosion and the method of producing it." The "explosion" probably took place in the printing plant, the editor furnishing the method.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

A Series of Grounds

By V. T. Kropidlowski

Two of our generators connected as in Fig. 1 and supplying current for shop lights and to a motor driving a locomotive transfer table, had been overheating considerably whenever the transfer table was used continuously for some time. One day generator No. 1 refused to carry its share of the load, and, moreover, when the motor was first started the pointer of the ammeter ran up even with that of the other machine, but then immediately dropped back to zero and remained there. I also noticed that the motor was taking more current than usual. From this it appeared that there was something wrong with the motor as well as the generator.

I decided to test the motor for grounds and having no special instruments for that purpose, used a 16-cp. lamp with sufficient cord to make the connections shown in Fig. 2. One end of a piece of wire was attached to one jaw of the main switch and the other end to the rail. One of the free ends of the lamp cord was connected to the other jaw of the switch, and with the other free end I began touching the contact rings of the controller, but no ground appeared. I then touched the contact fingers, and when on Nos. 1 and 2 the lamp lighted but burned brighter when touching finger No. 1. Of course, while this was being done the controller was in the off position, and the main switch was open.

Upon tracing the circuits from these fingers they proved to be the two armature leads. By a process of elimination, I continued disconnecting the leads from the brush-holders and touching the commutator segments, but the ground still showed. Upon removing the brushes from the holders and trying the commutator again no ground showed, but upon touching the brush-holders the ground again appeared. This led to an examination of the brush-holders and disclosed the fact that the nut and bolt securing one brush-holder to the frame was in contact with the motor casing—a good metal-to-metal contact.

Upon looking at the bolt of the second brush-holder I found it also grounded, but not a metal-to-metal contact, in which case a dead short-circuit would have existed. It developed that the brush-holder became warped, allowing No. 1 to come in contact with the frame, and at No. 2 dirt was lodged between the nut and the frame. By taking out the brush-holder frame and straightening it this trouble was eliminated.

Having the ground cleared at the motor, I returned to the engine room and found generator No. 1 still refusing to carry its load, although the motor was not exacting any more current than its load required. The machine acted as if its series field was not active, so I disconnected the series field terminals and passed a current through it, using the same incandescent lamp as that for locating the ground in the motor, but found that nothing was wrong there.

I then began looking for loose connections, and finally discovered that the negative cable of No. 1 machine could easily be pulled out of the connector where it connected with the fuse at the back of the switchboard (indicated at *C*, Fig. 1), the solder having melted out from around it. This accounts for the generator picking up its load at the instant the motor was started, but at the same instant the heavy current pulled away slightly the loose end in the socket and the effect was very much the same as opening the circuit by pulling its switch. After repairing this, the machine carried its share of the load as before.

But this was not all; another puzzle developed. After having apparently corrected everything, I decided to test the generators for grounds, using the same lamp for this purpose, and found a ground heavy enough to light the lamp brilliantly, although the frames were supposed to be insulated.

After trying for a week to locate this ground, I found it where the brush lead is soldered to the series field on No. 1 machine, as indicated at *A*, Fig. 1. There was a heavy lead thimble poured around the joint, but through carelessness in pouring, some of the lead had flowed onto the pole-piece, thus making a good contact with the frame. This had been covered by the tape and therefore had not been noticed before.

As to how the circuit was completed through the ground, when the machine was supposed to be insulated will be made clear by referring to Fig. 3. Water is used on the bearings of the machines, the piping being as shown, and the ground was evidently completed through

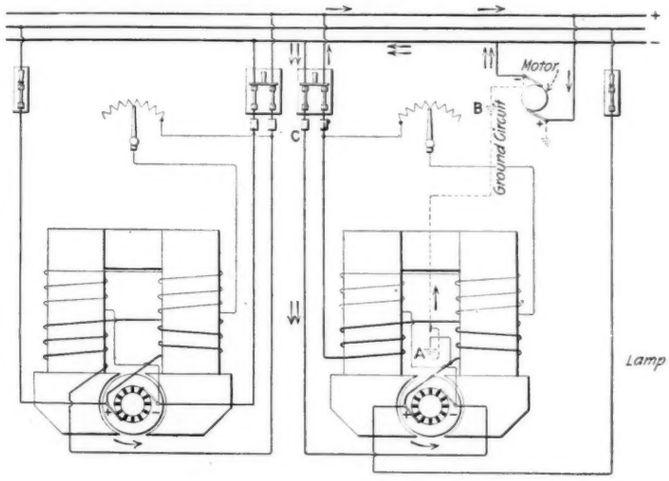


Fig. 1. Generator Circuits

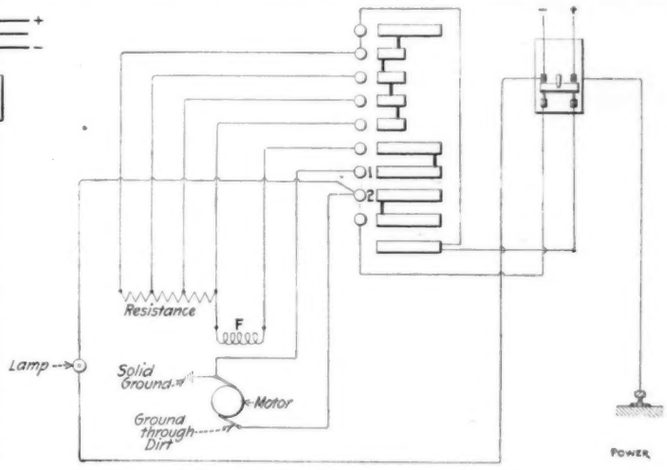


Fig. 2. Showing Method of Testing for Ground in Motor Circuit

the stream of water, the circuit being as indicated by the dotted lines.

The melting of the solder in the socket at *C*, Fig. 1, was due to the brush lead being grounded to the frame through the solder (indicated at *A*, Fig. 1), and to the grounds existing at the motor. The metal-to-metal contact with the ground existed at the negative brush of the motor, and the ground at the generator was at the positive brush. If it had not been for the ground current going through the stream of water (which had a comparatively high resistance) a dead short-circuit would have resulted and something would have burned out. As it was, however, there was a steady flow of current in addition to that supplied to the motor (which was heavier than normal, due to its double ground) but not enough to burn out anything in a short time. Eventually, however, it melted the solder in the connector of the negative lead.

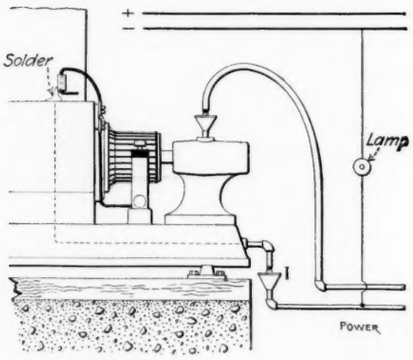


FIG. 3. SHOWING HOW GROUND WAS COMPLETED THROUGH SOLDER AND WATER

Referring to Fig. 1, the single arrows indicate the current in the positive leads and the double arrows the motor current plus the ground current, which did not go through the positive leads, but entered the ground at *A* and the negative leads at *B* at the grounded brush of the motor.

CORRESPONDENCE

Ground Caused Motor to Start

In Fig. 1 is shown the wiring diagram of the switchboard control for the steam turbine in the plant under consideration. It consists of a small electric motor having two field windings and mounted on the governor stand. Through the medium of a worm gear this motor varies the tension of the governor spring, thereby varying the steam admission and consequently the speed of the turbine. The two field windings are for running the motor in either direction—only one winding being in use at a time—and the direction of rotation is determined by the position of the single-pole double-throw switch shown.

This switch is on the switchboard and is normally held open by a spring at the motor. There are two limit switches which are opened automatically by the moving element of the control apparatus to prevent jamming the parts should the switch fail to open on the switchboard.

From the exciter busbars (to which the motor is connected) lights are also supplied about the station as indicated. One evening the motor suddenly started of its own accord and ran the moving element down; this opened the limit switch for that direction of rotation, but the motor kept running until the blade of the limit switch opened so wide that it jammed into the gear wheel of the control and stopped the motor. It was found in this position smoking hot with the current still passing through it.

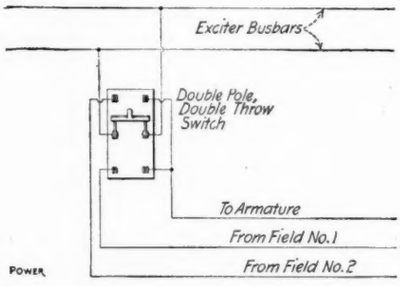


FIG. 2. WITH THESE CONNECTIONS THE TROUBLE COULD NOT HAVE OCCURRED

The fuses were immediately pulled to cut off the current, and an investigation made. This developed that a ground had occurred in one of the field windings and another on the negative side of one of the lighting circuits, thus furnishing a complete circuit which was not broken by the opening of the limit switch. The path of the current was from the ground in the field coil to the ground in the light circuit and back to the negative busbars of the exciter. This is plainly indicated in the sketch, Fig. 1.

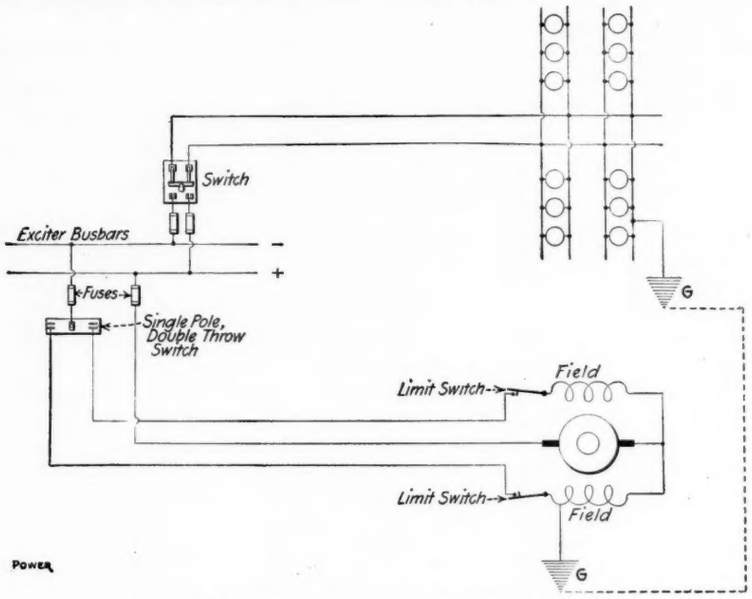


FIG. 1. CONNECTIONS WHICH CAUSED TROUBLE

Had the wiring of the motor been as in Fig. 2, the trouble would not have happened.

C. L. GREER.

League City, Texas.

What Made Transformers Heat?

In answer to C. J. Brindley's inquiry in the July 2 issue as to why the insulation melted on his current transformers when the secondary coils were open-circuited, I would offer the following explanation:

For measuring instruments, transformers are designed to have small magnetizing currents and small coil-flux densities; the magnetomotive force of the secondary when closed is nearly equal to that of the primary and opposed to it. When the secondary is opened the magnetomotive force due to it disappears and therefore the core has acting upon it a large unopposed magnetomotive force, which causes the core flux to increase, thereby increasing the secondary potential.

If the resistance of the secondary circuit is fixed, the secondary current will increase in proportion to the secondary voltage and hence in proportion to the current in the main circuit. This increased current will burn out a transformer if the secondary coils are left open-circuited when disconnected from the instruments.

W. C. FISHER.

Brooklyn, N. Y.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

American Practice in Rating Internal Combustion Engines—II*

By T. C. Ulbricht and C. E. Torrance, Jr.

Mechanical Efficiency

From a large number of values obtained from American manufacturers and operators, the following average table was derived for the mechanical efficiencies of engines operating on various fuels:

Producer Gas	82.0 per cent.
Natural Gas	84.0 per cent.
Illuminating Gas	84.2 per cent.
Gasoline	87.7 per cent.
Oils	84.8 per cent.

These averages are on the basis of fuel used, instead of the engine type and erage thermal efficiencies of American engines, letters were sent to about ninety of the largest manufactures in the United States, requesting guarantees on the brake horsepower, thermal efficiency on this basis, kind and calorific value of fuel upon which the guarantee was based, and variation of guarantee, if any, with the size of engine.

The thermal efficiencies were mostly calculated from the guaranteed fuel consumption at full load, by the formula,

$$\text{Thermal efficiency} = \frac{2545}{B.t.u.\ per\ b.hp.}$$

the numerator being the B.t.u. equivalent of one horsepower-hour.

The curves, Figs. 13 to 20, show the results as obtained in most cases from actual guarantees given by the manufacturers, and since the tendency seems to be to under-rate the engines, or place the guarantee on the safe side, it would seem that these average curves represent good practice and are too low rather than too high.

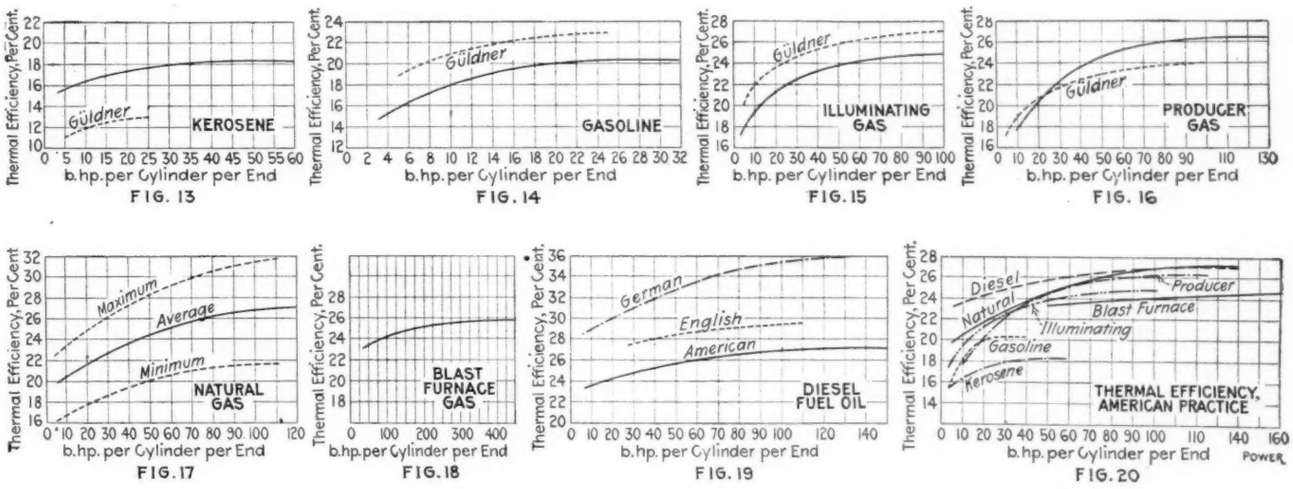
Fig. 13 shows the average thermal efficiency curve for engines using kerosene. Where possible, Güldner's values for German practice have been plotted on the same sheet with those representing American practice. For kerosene, Güldner's curve is found to be below the average for American practice. This discrepancy is probably due to the fact that when Güldner wrote, some ten years ago, very few oil engines had been developed, and the thermal efficiency was consequently low.

The thermal efficiency is seen to increase with an increasing brake-horsepower, approaching 18.4 per cent. above 40 b.hp. Fig. 14 shows the average thermal efficiencies for gasoline, which is lower than Güldner's curve for German practice. This would indicate either a higher development of the German gasoline engine, or a too conservative guarantee by the American builder. The efficiency is low for small horsepowers, but increases until above 22 b.hp., where it reaches a maximum value of 20.4 per cent. Güldner's value at 25 b.hp. is 23 per cent.

Fig. 15 also shows that for illuminating gas, Güldner's curve lies above that for American engines. At 100 b.hp., American practice shows 25 per cent., while German practice shows 27 per cent. For producer gas, American practice in general gives results higher than the German, as will be seen from Fig. 16.

As may be expected, due to the great variation in the analysis of natural gas, there is a wide range in the thermal efficiencies of engines using this fuel. (Fig. 19). Therefore the average, maximum and minimum curves are given. German practice does not include this gas. The average curve reaches a maximum of 27.3 per cent. at about 120 b.hp. Very few values were obtainable for blast-furnace gas, but the curve, Fig. 18, is what might be expected from good practice, showing a maximum of about 26 per cent. above 400 b.hp.

Special efforts were made to obtain values for the Diesel engine, and the result (Fig. 19) show the German values of thermal efficiency to be highest, the English somewhat lower and the American



Thermal Efficiency Plotted against Bbake Horsepower for Different Gases

in the case of gasoline and oils, are somewhat above the values usually obtained. However, the averages for the three gases are just what might be expected on any commercial test.

Thermal Efficiency

The thermal efficiency of a gas engine is rather indefinite, unless it is stated whether it is based on the work developed in the cylinder, or on that delivered at the brake. In this investigation all thermal efficiencies have been referred to the brake horsepower per cylinder per end, so that a builder or purchaser may know just what percent of the total heat units put into an engine is obtained at the brake as useful work.

To obtain data for determining the av-

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still lower. This may be accounted for both by the longer period of development of this engine abroad and the higher quality of workmanship in Germany than in other countries.

Fig. 20 shows all the preceding average curves, for American practice only, reduced to the same scale for comparison.

Thermal Efficiency Drop with Increase of Compression

In discussing this phase of the subject, it will be assumed that the mixture can be regulated and cooling effected so that preignition will not occur. With each increase of compression pressure, there is a corresponding increase of maximum explosion pressure. At some point this pressure becomes such that further increase is impossible, as the friction in the bearings, pins, etc., increases rapidly and places undue wear on the material. This increase in friction decreases the mechanical efficiency, which directly affects the brake horsepower, resulting in a decrease of thermal efficiency above a certain point.

Also, with an increase of compression and explosion pressure, the maximum temperature becomes excessive, resulting in a large and rapid loss of heat along the expansion line to the jackets, in radiation, etc., so that above a certain point, the thermal efficiency decreases with any increase in compression pressure.

In the theoretical Otto cycle, it can be shown from the efficiency forumla,

$$E = 1 - \frac{Pd}{Pa}$$

that as the compression pressure Pa increases, the fraction Pd/Pa becomes smaller, giving a constantly increasing efficiency. Actually, however, as already explained, practical considerations so limit this increase, that above a given point (determined by Güldner to be 200 lb. per square inch) the thermal efficiency actually decreases.

Volumetric Efficiency

The volumetric efficiency of a gas engine is the ratio of the actual volume of fuel mixture taken into the cylinder, to the piston displacement. It varies with the compression pressure, the piston speed, the suction resistance, the temperature during the suction stroke, etc.

It is usual in laboratory experiments to consider a "lower-loop" diagram, made by an indicator with a light spring, and actually measure the lengths a and b, shown in Fig. 21. Then the apparent volumetric efficiency is $a \div b$, but this does not take into account the temperature changes.

If a' is the volume of the charge at 14.7 lb. per square inch pressure and 62 deg. F. temperature, the actual volumetric efficiency is $a' \div b$. The method used in calculating the volumetric efficiency is illustrated in the following example for an actual engine using producer gas. Where the thermal efficiency was not given with the test, it was taken from the curves already given. In most cases, the correct ratio of fuel to air, as determined in the article on oils and gases (to appear later), were used in this calculation.

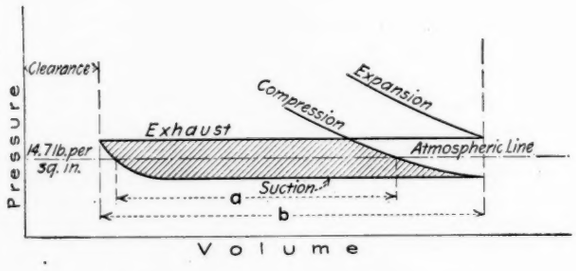


Fig. 21. "Lower Loop" Diagram

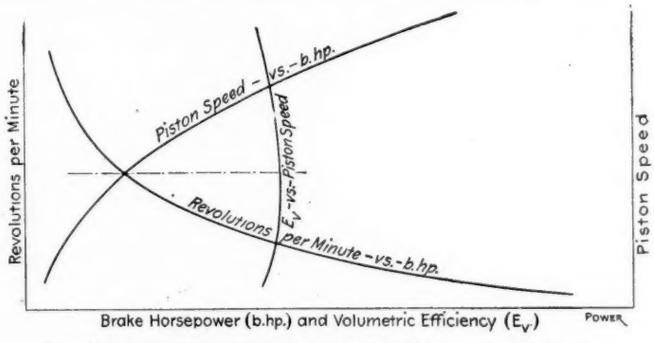

Revolutions per Minute
Piston Speed – vs. – b.hp.
Piston Speed
Revolutions per Minute – vs. – b.hp.
Piston Speed
Brake Horsepower (b.hp.) and Volumetric Efficiency (E_v)
Power

Fig. 23. Showing Point at which Maximum Charge Enters Cylinder

Example: The engine used for this determination was of the single-acting horizontal type, having a 21¼-in. cylinder, a 34½-in. stroke and developed 125 hp. at 145 r.p.m., corresponding to a piston speed of 834 ft. per minute. The following results were obtained:

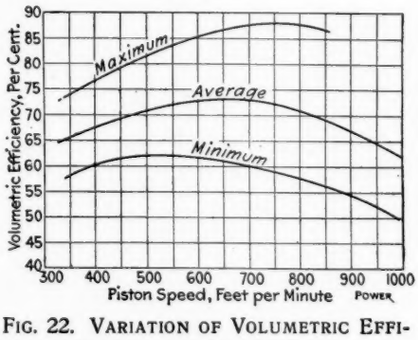


Fig. 22. Variation of Volumetric Efficiency with Piston Speed

Piston displacement = 7.08 *cu.ft.*

B.t.u. per hour = 2545 × *b.hp.* = 318,125

Cycles per minute = *r.p.m.* ÷ 2 = 72.5

$$B.t.u.\ per\ cycle = \frac{318{,}125}{72.5 \times 60} = 73.1$$

The thermal efficiency is given as 29 per cent.; hence each cycle requires in the cylinder

$$\frac{73.1}{0.29} = 252\ B.t.u.$$

One cubic foot of gas has an average heat value of 120 B.t.u., therefore the cubic feet of gas required are

$$252 \div 120 = 2.1\ cu.ft.$$

For producer gas the ratio of air to gas is from 1:1 to 1:1.25. With 1:1 there would be required 2.1 cu.ft. of air, giving a total mixture of 4.2 cu.ft. The volumetric efficiency is, therefore,

$$4.2 \div 7.08 = 52\ per\ cent.$$

With a ratio of air to gas of 1:1.25 there would be 2.63 cu.ft. of air required, making a total mixture of 4.73 cu.ft., and giving a volumetric efficiency of

$$4.73 \div 7.08 = 67\ per\ cent.$$

Fig. 22 shows the average, maximum and minimum values of volumetric efficiencies (E) plotted against piston speeds. The average curves show that engines of low piston speed have a low volumetric efficiency, but this increases with the piston speed until a maximum is reached at about 700 ft. per minute piston speed, after which it rapidly drops. This may be explained as follows:

For small engines of high rotative speed and short stroke in proportion to the cylinder diameter, the piston speed is low. The time for accelerating the charge volume is very short, resulting in a low final velocity. The curves plotted between cylinder diameter and valve diameter (to appear in a later article) show that they are proportional according to the following relation:

$d_v = 0.345\ d$ (*for the inlet valves*)

$d_v = 0.350\ d$ (*for the exhaust valves*)

d_v being the diameter of the valve and d the cylinder diameter.

Because of this, and the low final velocity and short time for acceleration, the charge volume is reduced, resulting in a correspondingly low volumetric efficiency.

On the other hand, as the sizes become large, the rotative speed is reduced, but the stroke becomes longer in proportion to the cylinder diameter, giving high piston speeds. The valve diameter is

proportional to the cylinder diameter in all sizes, but the piston speed has increased much more rapidly than the cylinder diameter, giving a very high maximum velocity to the incoming charge. The time for acceleration has been slightly increased, and the

However, in sizes intermediate between the limits cited, the volumetric efficiency is found to reach a maximum value. This is because in the intermediate sizes, there is a point where the rotative speed becomes proportional to the piston speeds (see Fig. 23), and

ciency is lowered, together with the developed horsepower.

Mean Effective Pressures

As the rating of a gas engine depends indirectly on the mean effective pressure, the accompanying mean effective pres-

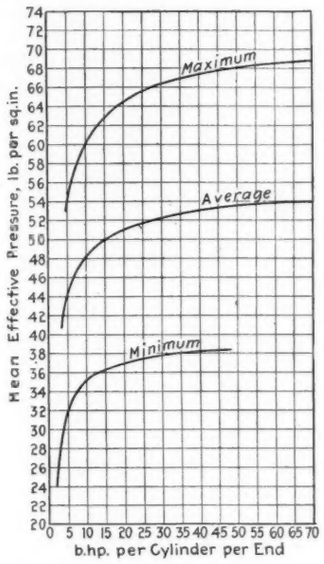


Fig. 24. Oils and Distillates

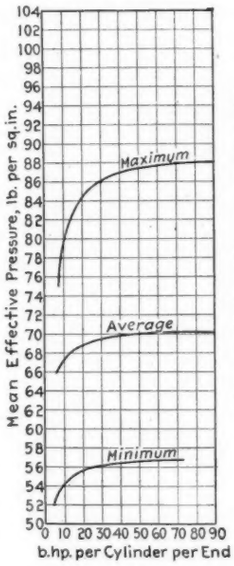


Fig. 25. Gasoline

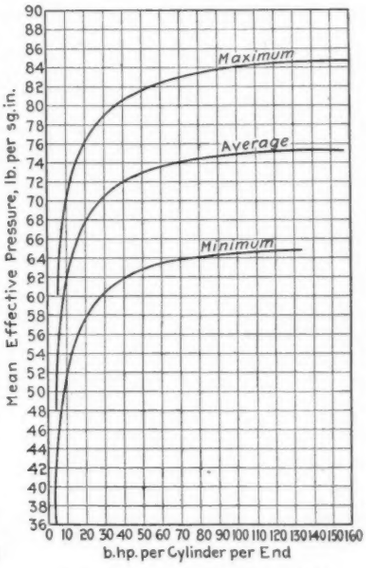


Fig. 26. Illuminating Gas

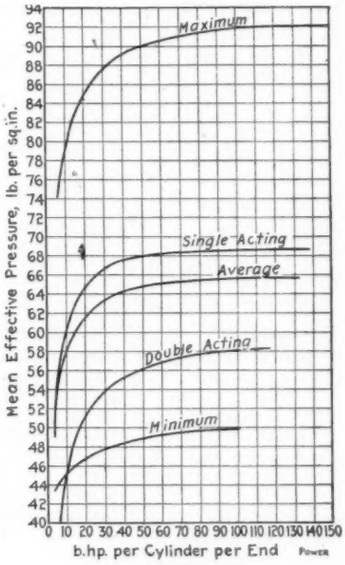


Fig. 27. Producer Gas

weights of gas entering a small and a large engine cylinder are proportional to two air columns, each of 1 sq.in. sectional area, which lengths represent the distances between the ends of the respective cylinders and their piston faces at the point of maximum acceleration for the small and large cylinders.

Because the valve diameter is still pro-

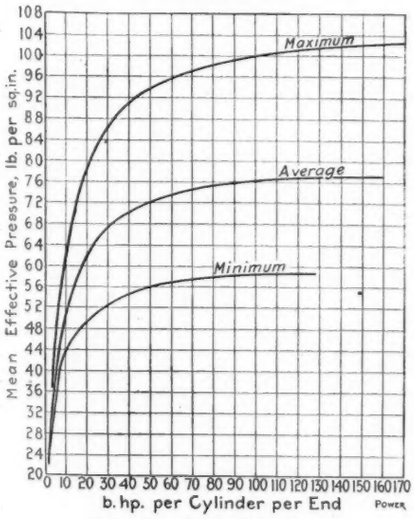


Fig. 28. Natural Gas

portional to the cylinder diameter, in all sizes, and also as the final velocity attained in the second case is very high, the air column, or charge volume is practically "wire drawn," or reduced in area, resulting in a charge volume which is too small in proportion to the cylinder volume and therefore giving a low volumetric efficiency.

therefore proportional to the cylinder volume. In this field, therefore, the velocity of the charge volume and the time allowed to acquire this velocity have just become balanced, allowing a maximum amount of charge to enter the cylinder, which results in a maximum volumetric efficiency. The volumetric efficiencies, as determined for American practice, are found to be somewhat lower than Güldner's values for German practice.

GULDNER'S VALUES OF VOLUMETRIC EFFICIENCIES

1. Slow speed, inlet valve mechanically operated	88 to 93 per cent.
2. Slow speed, automatic inlet valves	80 to 87 " "
3. High speed, inlet valves mechanically operated	78 to 85 " "
4. High speed, automatic inlet valves	65 to 75 " "
5. Very high speed automobile engines, with automatic inlet valves and air cooling	50 to 65 " "

This may be accounted for in several ways, chief of which is the universal tendency of American manufacturers to under-rate their engines. A careful study of the data received, shows that a majority of the engines will carry from 10 to 30 per cent. overload, which would mean a higher volumetric efficiency.

Another possible reason for the large variation between maximum and minimum values, is that many manufacturers fail to state the elevation for which a given engine will develop a given horsepower. This is important in the proper rating of gas engines, because at a high altitude an engine will not develop as much power as at sea level, due to the difference in pressures. Hence, the charge volume at the high elevation is less, and therefore the volumetric efficiency curves have been calculated and plotted between brake horsepower per cylinder per end, as abscissae and mean effective pressure in pounds per square inch as ordinates.

The mean effective pressure was found from the following formula:

$$I.hp. = \frac{plax}{33{,}000 \times 12}$$

in which

$I.hp.$ = Indicated horsepower per cylinder per end;
p = Mean effective pressure;
l = Length of stroke in inches;
a = Area of piston in square inches;
x = Number of explosions per minute.

Representing the cylinder diameter by d and the revolutions per minute by n,

$$a = \frac{\pi d^2}{4} \text{ and } x = \frac{n}{2}$$

Substituting these values

$$I.hp. = \frac{p \pi d^2}{4} \frac{n}{2} \frac{l}{12} = \frac{p d^2 ln}{1{,}010{,}000}$$

Solving for p,

$$p = \frac{1{,}010{,}000\ i.hp.}{d^2 ln}$$

Having already determined the mechanical efficiency for the different cases, the relation between indicated horsepower and d^2ln were found from the curves plotted between $b.hp.$ and d^2ln (Article I). Using this relation in connection with the foregoing formula, curves were plotted between mean effective pressure and brake horsepower per cylinder per end. These are shown in Figs. 24 to 28, inclusive.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Means for Accumulating Refrigeration—I

By Charles H. Herter

This article deals with means available for accumulating or storing up refrigeration, particularly in cold-storage rooms where the temperature must be kept as nearly uniform as possible. Reserve refrigerating capacity allows operating intermittently and provides for the contingency of accidental shutdowns, which is an advantage in plants equipped with but one machine.

In "Relative Costs of Continuously and Intermittently Operated Refrigerating Plants," in the issue of Sept. 19, 1911, the writer showed that, as a rule, plants with a capacity of 10 and more tons of refrigeration per 24 hours are most economical in yearly operating expense when run continuously in summer. Some of the following remarks apply especially to such plants. In the winter intermittent refrigeration is quite common, the low temperature outdoors making 24-hour operation unnecessary. In certain localities electric power may be had at favorable rates between peaks at the station load, and there intermittent refrigeration even for larger plants might prove as economical as continuous 24-hour operation. Labor, power cost, size of equipment, life of plant, kind and quantity of goods stored, etc., differ greatly with various installations, and it is therefore not possible to cover all conditions in a brief article.

The temperature rises in a cold-storage room, when refrigeration is shut off, mainly by heat penetrating the inclosing walls. Goods not yet cooled to the room temperature will also raise the temperature, but to simplify the calculations this will be ignored for the present. If the walls, floor and ceiling are not well insulated or the doors do not close tight, heat may get in so fast that the desired temperature can be maintained approximately uniform only by refrigerating continuously. Therefore, close attention must be paid to the fit of the doors, windows and other openings to see that they close air-tight with little effort.

A high-sill door is kept tight more easily than one with a low sill, because the joint is continuous around the opening and the high sill is also free from wear due to trucking. Some refrigerator-door manufacturers make a flexible gasket which is subject to great wear, and, as the cold, heavy air seeks to flow out at the bottom this source of leakage should receive constant attention. To test for air leakage, blow tobacco smoke against the cold side of the closed door; if any comes through the crevices the door should be refitted. A mistake frequently made is ordering doors unnecessarily large, because they cost little more than smaller ones, whereas the latter are more easily kept tight, displace less square feet of wall insulation, and allow less loss of cold air when the door is opened.

Windows in cold-storage rooms are not used by those aware of their low insulating value. For a temperature of 32 deg. F. a cold-storage room should be so insulated that the heat flow from the outside to the inside does not exceed 2 B.t.u. per square foot per 24 hours per degree Fahrenheit air temperature difference. Using two thicknesses of glass inclosing one air space for a window, the heat transmission coefficient is 13 B.t.u., while even with four thicknesses and three air spaces it is 6 B.t.u., with air-tight windows. Glass containing 1 to 2 per cent. of iron oxide, imparting a greenish tint, is preferable as it passes no radiant heat.

The amount of material stored also affects the maintenance of temperature. A meat room 20x40x15 ft. high, aggregating 12,000 cu.ft., can hold 50,000 lb. of meat. If the four walls are brick, 16 in. thick, insulated inside with ½ in. of mortar, 3 in. of corkboard, and ½ in. of portland cement finish, making a total thickness of 20 in., the heat-transmission coefficient is about 2 B.t.u. If the floor, built directly upon the ground, is 6 in. of concrete, 3 in. of corkboard coated bottom and top with hot asphalt, and a covering of 3 in. of concrete wearing floor, the heat-transmission coefficient is 2.3 B.t.u. If the ceiling (not a roof) is 6 in. of concrete with the insulation laid on top where its application is cheaper than underneath, the construction to be the same as for the floor, the heat loss will be 2.2 B.t.u. Thus, the heat entering this well insulated room per 24 hours will be

	Sq.ft.	Deg.		B.t.u.
Walls,	1800	(80–32)	×2	172,800
Floor,	800	(60–32)	×2.3	51,520
Ceiling,	800	(75–32)	×2.2	75,680
Total				300,000

equivalent to 12,500 B.t.u. per hour.

In large and important installations the heat leakage for the different walls should be calculated separately according to the method used by heating and ventilating engineers and a higher outside temperature should be assumed on the sunny side than on the north side, also allowances made for wind exposure, etc., but with an imaginary building such refinements are not necessary to the subject in hand. For the same reason, the varying amounts of refrigeration for day and night need not be considered, it being sufficiently accurate to assume mean temperatures to be constant during 24 hours.

Chilled meat is stored at between 31 and 36 deg. F., depending upon its kind, the degree of humidity, the duration of storage and individual opinion as to how it should look and taste. Siebel, in his "Compend of Mechanical Refrigeration," says: "Meat, kept constantly at 31 deg. F., can be kept fresh six months, but if allowed at times to go as high as even 33 deg. is likely to spoil in less than a month; however, if the ventilation and humidity are properly regulated it should keep about two months in good condition in the latter case." By operating intermittently and with the air temperature never below 32 deg., actual freezing of the meat is avoided and refrigeration is not burdened with the latent heat of freezing.

The foregoing 300,000 B.t.u. of heat will raise the temperature of the 50,000 lb. of meat, after 24 hours' shutdown

$$\frac{300{,}000\ B.t.u.}{50{,}000\ lb. \times 0.8\ spec.\ heat} = 7.5\ deg.$$

or 3 deg. in 9.6 hours. If the room contains only 25,000 lb., the meat temperature will rise 3 deg. after a stop of only 4.8 hours. These figures are merely approximations; in reality, much of the heat, as shown later, is retained by the walls, and little or none reaches the inner portions of the cooled meat. Freshly stored carcases with the inner temperature still above that of the room will appropriate the available refrigeration until the temperatures are equalized. By considering the total square feet of surface of the relatively warm meat and the coefficients of radiation and convection per degree of average temperature difference obtaining between meat surface and surrounding air during the time limit set, this quantity of heat may be determined.

As the foregoing shows, food stored in a well filled room will suffer less during a shutdown than when stored in a room practically empty; also the amount of insulation influences the temperature rise

during a shutdown. If, as is often true, the heat leakage averages 4 instead of 2 B.t.u., 600,000 B.t.u. will enter the room in 24 hours, not only increasing the work of the refrigerating machine but also causing a 3-deg. temperature rise to take place after 2.4 hours' stopping. The cold air confined in the room need not be included in these figures because its capacity for heat is only 0.24 B.t.u. per pound per degree, only

$$12{,}000 \times 0.0807 \times 0.24 = 232 \ B.t.u.$$

being required to warm up all the air in the room 1 deg. when empty. With material in storage the amount of air will be much less.

Before calculating the heat the walls can absorb, several other factors may be considered which help to maintain temperatures; for instance, the frost collecting on the cooling pipes. On 1000 lineal feet of 1¼-in. pipe coated ¼ in. thick with snow weighing approximately 7 lb. per cubic foot, there would be 73 lb. of snow, its sensible and latent heat not amounting to more than 144 B.t.u. per pound, or a total of 10,512 B.t.u. For completeness might be included the heat absorbed by the metal of the 1¼-in. extra-heavy cooling pipes when warming from, say, 12 deg. to 32 deg., this being

$$3000 \ lb. \times 20 \ deg. \times 0.116 = 6960 \ B.t.u.$$

The total for snow and pipes would be 17,472 B.t.u., an amount sufficient to lengthen the period of shutdown 1.4 hours.

The shut-down time can be longer when brine instead of ammonia is used in the cooling pipes. With the brine system the average temperature difference between circulating brine and air may be 16 deg., necessitating the use of

$$\frac{1000 \ ft. \times 20 \ deg.}{16 \ deg.} = 1250 \ ft.$$

of 1¼-in. extra-heavy brine pipe, to do the same work as 1000 ft. of 1¼-in. direct-expansion piping. This piping when filled contains 11 cu.ft. of brine. The heat capacity of a cubic foot of salt brine per degree Fahrenheit is about 58 B.t.u., and of chloride of calcium brine, 52 B.t.u. Using the latter figure, the 11 cu.ft. in rising 16 deg. absorbs

$$11 \times 52 \times 16 = 9152 \ B.t.u.$$

Contrary to general opinion, the brine in the pipes delays the 3-deg. air temperature rise by only

$$\frac{9152 \times 24}{300{,}000} = 0.73 \ hour$$

Altogether, then, the plant can be shut down 6.93 hours and not rise more than 3 deg. in temperature.

	Hours
25,000 lb. of meat	4.8
Snow and pipes	1.4
Calcium brine	0.73
Total	6.93

If the insulation used allows heat to enter twice as fast, the air temperature will be up to 35 deg. after a stop of only

$$\frac{6.93}{2} = 3.46 \ hours$$

and if the room contains only 5000 lb. of meat, the stopping time dwindles to a total of 1.55 hours.

Good insulation not only permits operating the plant with the least refrigeration, power, time and cost, but also helps to reduce fluctuations in room temperature. After stopping refrigeration the inflow of heat continues, although at a constantly decreasing rate, until after days the inner air temperature will be within a few degrees of the atmospheric temperature.

The stopping period of 6.93 hours just calculated, representing a heat influx of 86,624 B.t.u., is further affected by the temperature variations of the walls, floor and ceiling. These in turn are the effect of temperature changes taking place outdoors. Hence before proceeding, a table giving the hourly temperatures outdoors must be decided upon as a basis. At the December, 1896, meeting of the American Society of Mechanical Engineers, F. H. Boyer read a paper containing an engineer's log of a large meat-packing plant at Boston, Mass. In this are recorded the outdoor air temperatures for July 18, 1896, the average for the 24 hours being 69 deg. F. An average of 80 deg., the temperature used herein, will represent the true condition for a larger territory than 69 deg. Boyer's hourly readings must therefore be raised about 11 deg. Then the hourly temperatures will be as in Table 1.

TABLE 1. HOURLY TEMPERATURES

Increasing Temperature Period	Deg.	Decreasing Temperature Period	Deg.
5 a.m.	70	2 p.m.	89
6 a.m.	73	3 p.m.	87
7 a.m.	77	4 p.m.	85
8 a.m.	80	5 p.m.	83
9 a.m.	82	6 p.m.	81
10 a.m.	84	7 p.m.	80
11 a.m.	86	8 p.m.	79
12 a.m.	88	9 p.m.	78
1 p.m.	90	10 p.m.	77
Average	81.1	11 p.m.	76
		12 p.m.	75
Average temperature for 24 hours, 79.6 deg.		1 a.m.	74
		2 a.m.	73
		3 a.m.	72
		4 a.m.	71
		Average	78.6

Table 1 shows that the lowest temperature obtains at 5 a.m. and the highest at 1 p.m., thus there is an increasing temperature period of eight hours and a decreasing temperature period of 16 hours. To add to the complication, at night the room temperature rises and the outdoor temperature falls, at which time the rate of heat flow will be correspondingly less per hour than during the day when the conditions are reversed.

To get an insight into these temperature fluctuations it will be sufficient to examine the four insulated walls and assume that these contribute 57,800 B.t.u., or three-fourths, toward the total 86,624 B.t.u. allowed to enter before the permissible 3-deg. temperature rise has taken place in all contents of the room. To analyze all surfaces separately is hardly justified here, particularly as the amount of goods stored, their initial temperature, the actual atmospheric temperature, and many other factors are so variable as to upset any fine calculations.

To determine the heat conducted through a solid wall of any material, the following formula can be used:

$$B.t.u. \ conducted \ per \ hr. = \frac{C}{Thickness \ in \ in.}$$

where C is the coefficient of conductivity, or the heat conducted through 1 in. thickness, in heat units per square foot per degree (Fahrenheit) difference in wall temperatures per hour.

The coefficient of conductivity of brick with a specific gravity of 1.98, or 123.5 lb. per cubic foot, is, according to Peclet, 5.56 B.t.u. Then for a 16-in. brick wall the heat transmitted per degree difference of temperature between the two wall faces is

$$\frac{5.56}{16} = 0.3475 \ B.t.u.$$

per square foot per hour. If this wall, as in the meat room under investigation, is covered with insulating material—namely, with a ½-in. layer of cement mortar, 3 in. of corkboard, and a ½-in. finishing coat of portland cement plaster, total thickness 20 in.—the heat conducted from the outer to the inner face per degree wall temperature difference can be calculated according to the following formula:

$$B.t.u. \ per \ sq.ft. \ per \ hr. = \frac{1}{\left(\frac{d}{c} + \frac{d_1}{c_1} + \frac{d_2}{c_2} + \frac{d_3}{c_3}\right)}$$

where d, d_1, d_2 and d_3 indicate the respective thicknesses and c, c_1, c_2 and c_3 the corresponding coefficients of conductivity.

For fine plaster of 1.25 specific gravity (78 lb. per cubic foot), Peclet gives $c = 4.2$ B.t.u. per hour for 1 in. thickness. The specific gravity should always be stated because the conductivity varies with the density. It varies also with the temperature, but not sufficiently to affect these calculations. The heat transmission through some good grades of American compressed corkboard 2 in. thick varies, according to recent tests, from 3.2 to 3.5 B.t.u., depending upon the quality or make, per square foot per 24 hours per degree difference in temperature between the air on the two sides.

From personal research, the writer has found that he must distinguish between the amount of heat passed through a test piece per degree air temperature difference and that passed through per degree wall-surface temperature difference. The necessity for such distinction has never been emphasized; nevertheless, so long as radiation and convection are to be reckoned with, there must be a difference or drop in temperature between a wall sur-

face and the surrounding air. To calculate the heat passing through a compound wall according to the accepted formula, the conductivity must be known of each separate layer as referred to its wall-temperature difference.

With practically still air the heat radiated per square foot per degree difference per 24 hours by smooth steam-heating pipes is 30 B.t.u. The radiation from a corkboard up to a certain thickness at least will not differ materially from this value, for H. Recknagel, by actual tests, found it to be correct for rough stone walls. Under constant conditions as much heat as enters a wall, passes through and leaves it. Accordingly the following formula applies:

$$H = \frac{c}{d}(t_2 \quad t_3) = a_1 (t_1 - t_4) = a_2 (t_3 - t_4)$$

where

H = B.t.u.;
c = Coefficient of inner heat conductivity (per degree wall-temperature difference);
d = Thickness of wall (over all);
a_1 = Total coefficient of radiation (including convection) at the warm side of the wall;
a_2 = Total coefficient of radiation (including convection) at the cold side of the wall;
t_1 = Temperature of warm air;
t_2 = Temperature of warm wall surface;
t_3 = Temperature of cold wall surface;
t_4 = Temperature of cold air.

An analytical treatment of conditions met with in testing insulating materials is not really pertinent to the subject of this article; still, as some readers may desire additional information regarding the effect air agitation has upon K, the coefficient per degree air-temperature difference $t_1 - t_4$, this subject will be discussed in a later article.

Troubles in an Absorption System and Remedies—I

By W. S. Luckenbach

As the season for producing ice and refrigeration to the full capacity is now upon us, every man in charge of an ice plant needs to be resourceful to maintain it at full capacity. The writer, therefore, believes the following comments from his experience will be timely. Some apply to both the absorption and compression system, but most of them to the absorption system.

First in importance is a full charge of aqua ammonia at not less than 26 deg. density, 28 is better, if the ammonia pump is in first-class order, as less heat is required to liberate the gas. It is essential to notice at what pressure steam in the generator will do the best work, viz., to produce the most cold with the least fuel consumption, because, the lower the steam pressure giving the desired results the less cooling water must be pumped and consequently less fuel consumed.

To get best results the gas liberated in the generator should be not less than 14 per cent., and by referring to the standard table we find that 25.7 deg. (Baumé scale) aqua ammonia contains 28 per cent. ammonia and at 18 deg. 14 per cent.; hence with the rich liquor at, say, 26 deg. and the poor liquor at 18 deg., the proper distillation is accomplished; should, however, the rich liquor be 24 deg., then the poor should be 16 deg., to obtain the 14 per cent. of gas. This, however, can only be accomplished from the same amount pumped into the generator by the application of more steam heat. Hence the importance of maintaining a high standard of aqua.

It may happen that on one day the rich liquor is found to be 26 deg., and the poor liquor 18 deg., while the next day the rich liquor is 24 deg. and the poor liquor 20 deg. It will puzzle some as to what caused the change and what is the remedy.

This change can only take place in the exchanger. The hot poor liquor passes from the generator through the exchanger in one direction and the comparatively cool rich liquor passes in the other direction, thereby exchanging temperatures, and if one of the pipes, or coils leaks, the density of the poor and rich liquor will change. The remedy is to make a bypass and cut the exchanger out at once, and have the pipes removed as speedily as possible.

Sometimes the liquor in the machine gets low, while running in the ordinary way, and, after adding more ammonia, gets low again in a few days, with no leaks of any kind. This may occur when charging or recharging and distilling; considerable air enters the system and will occupy space in the condensing and cooling coils, finally finding its way to the absorber from which it is purged. It, however, frequently happens that the liquor lies dormant in the bath coils; especially may this be the case if the bath in the freezing tank is on a level with the machine room, instead of being above the liquor level in the absorber, thereby establishing a gravity system. The first move is to produce a partial vacuum so that any liquor lying in the bath coils will flow to the absorber. To do this, the liquor in the generator should be allowed to get low and the liquor in the absorber correspondingly high, then the poor liquor valve should be closed at the absorber and the generator filled as full as possible to avoid a boil-over. This will lower the liquor in the absorber and produce a partial vacuum.

CORRESPONDENCE

Brine Leak in Ice Tank

It is expensive to repair leaks in an iron ice tank when the tank has to be jacked up in order to get under it to drive the rivets. In fact, while letting the tank back into position, other rivets are loosened and new leaks are made. I stopped the leaks in a 30x40-ft. iron ice tank in a much quicker and less expensive manner.

The brine was pumped out of the tank into ice cans and the coils and tank were thoroughly washed. The coils were then set over to one side, leaving half of the tank to be worked on. This half was again washed and mopped thoroughly dry; all the seams and rivets were heated with a blow torch and brushed while hot with a stiff fine wire brush until all the scale and dirt were removed, leaving a bright surface. The seams and loose rivets were calked, using air. Where the heads of the loose rivets would not stand calking, the rivet was driven out and a tap rivet put in. The seams and rivets were then heated as before, and while hot they were painted with a mixture of hot coal-tar pitch and Ferrubron paint. This mixture was liberally applied, spreading the paint 4 or 5 in. on each side of the lap of the seam.

In making this mixture 75 lb. of Ferrubron paint was used to the barrel (300 lb.) of pitch; this paint gives elasticity to the pitch and thus tends to prevent cracking. The other side of the tank was treated in a like manner. This tank has been in service for two seasons, and, so far, has shown no sign of leak.

Gay A. Robertson.

Louisville, Ky.

Wear in Ammonia-Compressor Cylinder

I would like to offer the following as a possible explanation why horizontal ammonia-compressor cylinders wear more at the top than on the bottom surface of the cylinder, as related by Mr. Anderson in the June 18 issue. When a compressor is fitted with a spherical piston head the pressure tends to centralize the piston, the wet gases, and oil coming in on the rod will flow to the bottom surface of the cylinder and lubricate it, while the top surface remains comparatively dry, and consequently will show wear before other parts.

As stated, this condition applies to the spherical piston. Will Mr. Anderson give us his experience with the flat piston head, that is, do both types show the same wear?

J. H. O'Hern.

Denver, Colo.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.

Monadnock Bldg., Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

FRED R. LOW, Editor.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $5 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 21 Shillings.

Entered as second class matter, December 20, 1910, at the post office at New York, New York, under the Act of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT
Of this issue, 29,500 copies are printed.

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

Contents

Page

Compression in Engine Cylinders..... 140
Fan Engine Room Signals..... 143
Hydro-electric Development near Pittsburgh... 143
Large Steam Turbine Development..... 144
Explosion of Air Receiver Due to Oil..... 147
"Cino" Iron Body Gate Valve..... 149
Goetze Compound Metallic Packing..... 149
Albany Sectional Dumping Grate..... 149
The Baumé Hydrometer..... 149
A Series of Grounds..... 151
Ground Caused Motor to Start..... 152
What Made Transformer Heat?..... 152
American Practice in Rating Internal Combustion Engines. II..... 153
Means for Accumulating Refrigeration—I..... 156
Troubles in an Absorption System and Remedies—I..... 158
Brine Leak in Ice Tank..... 158
Wear in Ammoni aCompressoreCylinder..... 158
Editorials.....159–160

Readers with Something to Say:
Valve Seat Repair....Damaged Steam Pipe Due to Faulty Design....Stopping Leaky Rubber Diaphragms....Firebrick Walls Prevent Smoke....Taking Care of Drips....Badly Bagged Boiler....Blowing Safety Valves Waste Steam....Feed Pump Troubles and Repairs.....161–163

Questions Before the House:
Water in the Low Pressure Cylinder....Improving a Neglected Plant....Emergency Expansion Bolt....Lifting a Broken Plunger....Putting in Crankpins....Unnecessary Risks....How the Cylinder is Cut....That Red Hot Blowoff Pipe....Flue Gas Analysis. 164–166

Inquiries of General Interest..... 167
Study Questions..... 168

POWER

Temperance

In our general campaign for the welfare of those engaged in the field which we serve, we have felt that we could not ignore the importance of temperance in the use of intoxicants—hence the foreword in this issue. It is one of POWER's functions to point directions in which the engineer may improve himself or his conditions. Another is to suggest ways for increasing the efficiency of the plant, which naturally depends in a great measure upon the efficiency of the operator and that is never at its highest if he is a heavy drinker.

Two things that brought the subject of temperance prominently to mind at this time are alluded to in the foreword, a wreck which occurred on the Lackawanna R.R., July 4, in which 46 were killed, and the engineer was alleged to be under the influence of liquor, and the statements made by John L. Sullivan in a recent interview, concerning his career of hard drinking. His advice to others does not stop with temperance, for he recommends leaving liquor entirely alone. He claims that among all the men he knew, which was a very great number, he could not recall more than a dozen abstemious drinkers.

Time may have been, as some have claimed, when a man was not considered a gentleman unless he drank to excess, but happily that day has long since passed. Could we now but banish the treating habit we would remove the most subtle evil of the whole thing. In the several social clubs that have prohibited treating, drunkeness is now unknown.

Still, even with treating persisting, the strong minded man can refuse alcoholic drinks when he has had as much as is good for him, and take a cigar or a soft drink. Those who always do this can do it with better grace than formerly, for the tendency to ridicule the man who has back-bone enough to stick to his principles, or who knows what is best for him, is steadily growing less. Even habitual drinkers very generally now respect a man's desire to stay "on the water-wagon."

Where safety to human life requires temperance, no further argument is necessary. All railroads now require it, and more and more are other industries, if not demanding, at least putting a premium on men for responsible positions who do not over-indulge.

Boiler Accident Prevention

The stays in the dome of one of two boilers, connected together by a line containing a slip expansion joint having no studs and check nuts, were to be removed from a boiler in a brewery in Nanaimo, British Columbia, recently. One of the boilers was under pressure at the time of removing the bolts from the flange connection at the idle boiler. When the engineer removed the last bolt from the flange the pressure separated the expansion joint and the escaping steam killed him.

If the facts are as stated it would seem that the engineer was either careless or thoughtless. While the last few bolts were being removed he should have noticed the strain on them caused by the separating of the expansion joint. Surely a heavy blow on the last bolt must have been required to remove it. It is pitiable that he did not heed the warning.

The government inspector of boilers and machinery testified that he inspected the plant nine months previous to the accident. We understand that the laws of British Columbia require studs and check-nuts on such joints, but the inspector considered it safe and did not compel their use. Even though he acted in good faith, he is, in a measure responsible according to our understanding of the law.

Furthermore we cannot undestand how a competent jury could do no more than recommend a more rigid enforcement of the law as stated in *The Nanaimo Free Press* of June 25.

Equally responsible are the engineer in charge of the plant and the man who erected the faulty joint, if the owners did not refuse to equip the joint with

studs and check-nuts. This case is one of many and should serve to caution those having to do with boilers.

A boiler inspector's duties impose such grave responsibilities that they should not be entrusted to any who are not extremely cautious and thorough. Besides extensive knowledge of pressures, strength of materials, etc., a boiler inspector should have good judgment and an unrelenting vigilance upon which so much depends in safety to life and property.

Because a condition is permitted by law does not prove that it is safe. An inanimate license or certificate hanging on a wall will not prevent a disastrous flywheel explosion if an engine governor has been grossly neglected by the holder of the license. A license is indicative of good character, competency and scrupulous care on the part of the holder and he should aim to deserve its endorsement.

No amount of technical knowledge means much if unaccompanied by common sense. The Nanaimo accident and countless others are due to a deplorable lack of it.

Gasoline and Kerosene Supply

The oil companies seem to have taken advantage of the increased demand for gasoline, occasioned by the large number of motor vehicles and boats, by abnormally boosting the price. This is particularly noticeable in certain cities where the price has advanced 25 per cent. over that of last year.

In defense of their action the dealers and producers argue that the demand is fast exceeding the supply. Unfortunately the figures of the present year, with which to refute or substantiate this claim, are not yet available. However, while this economic theory may suffice as an explanation in the absence of other proof, it must be admitted that it costs no more to produce a gallon of gasoline now than it did a year ago; in fact, if we are to consider the usual effect of producing in quantities, it should cost less. Hence, it would appear that some are reaping enormous profits.

Let us consider how these conditions affect stationary practice, which is concerned not so much with gasoline as with the other products of crude petroleum. For every gallon of gasoline there is produced two or more gallons of kerosene and considerable fuel oil. We are told that the oil companies regard the kerosene as a byproduct and are shipping large quantities to foreign markets, where it is disposed of at a low figure, over twenty million barrels being exported last year alone.

Why not keep more of this kerosene at home and utilize it for power purposes in oil engines? Surely it would be more profitable to sell this to home trade, if a market could be created, than to compete in foreign markets, with the additional cost of shipping.

The "Principe di Piedmonte" Accident

The accident on the Italian liner "Principe di Piedmonte," an account of which appears on page 172, is of much concern to all steam users. This accident and that on the "North Dakota" in 1909 are identical, and many similar ones have happened in stationary practice.

These cases well illustrate the importance of non-return stop valves. Had the boilers of these vessels, and those in stationary practice where tubes have let go, been provided with them, the results would have been less serious. Even with several exits in a boiler room, and in most of them there are not, the chances of escape at such times are small enough, but to be entrapped in a space quickly filled with scalding steam is horrible beyond imagination.

Deep in the consciousness of every power-plant man, however competent he may be, there lurks the fear of a boiler under ordinary working pressures, and though he has never experienced a serious burst, a broken gage glass is indicative of what is possible.

However bad it may be when the steam from one boiler is suddenly liberated in a room, it is far worse when several connected to a common header belch steam and water that flashes into steam at the extreme pressure reduction. The fires cannot be drawn, and fortunately there is no need of it, as the dying fires would not produce enough heat to evaporate the water left in the boilers. A man's first thought, on such an occasion, is justifiably his own preservation.

It does not cost much to equip boilers or supply pipes with non-return valves or other means to prevent a discharge from all the boilers when only one is affected. The accidents that have occurred where no such provisions are made should be a warning to those building new plants or operating old ones. Desire to safeguard human life should be strong enough to bring about these reforms voluntarily without waiting for law to make them compulsory.

More Publicity Needed

Usually, when a central station succeeds in taking over an isolated plant, it is announced through the daily press, or literature is mailed to the business man who may be a possible purchaser. The central station, knowing the commercial value of publicity, seldom misses a chance to advertise its service. The isolated plant needs more publicity.

It is pertinently urged by *The Isolated Plant* that the manufacturer also address himself directly to the business man if he would hope to induce him to buy power-plant equipment and generate current instead of buying it. Now, he appears to be playing into the hands of the central station.

Very often home-made power would be cheaper than purchased current, but a campaign of education and publicity is badly needed to make known under what conditions the isolated plant is to be preferred. The business man listens to the louder voice, and at present the central-station people are making all the noise.

A wiseacre has recently gathered figures, according to *Harper's Weekly*, in which he attempts to show that in firing one ton of coal an hour (eight tons a day) a fireman expends but little more energy than the young-girl typist in operating a typewriter that length of time. He expends about 20,000 ft.-lb., she, 15,000 ft.-lb. He must add the shovel's weight to get his figure. She must produce 12,000 more or less cruel words to get hers. He (meaning the statistician this time) will probably "get his" if he starts this line of argument with a fireman.

Has the site of the original power plant been discovered? A British engineer claims that the Garden of Eden was located in Egypt, and it is proposed to restore the home of Chief Adam to its pristine glory. What a chance this offers to the efficiency engineer!

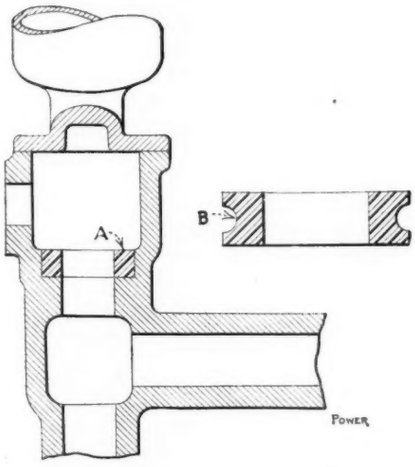
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Valve Seat Repair

The sketch shows one of the discharge chambers of a high-pressure pump, with the steel valve seat *A* in place. This pump works against 1500 lb. pressure on an accumulator for hydraulic work. The seat, which is supposed to be a press fit, came loose, allowing the fluid to pass between it and the cylinder walls, soon enlarging the walls.

Examination showed no alternative but to take the cylinder off, rebore and fit



VALVE CHAMBER AND STEEL VALVE SEAT

it with a new seat, valve, etc. As we could not spare the pump long enough for such a job, I took the seat out and turned the edge off, as shown at *B*. The cylinder was then heated to dispel all moisture and expand it as much as possible. After replacing the seat, the space between it and the cylinder wall was filled with good, hard babbitt and then well calked to make it tight. This job was done two years ago and the seat has shown no signs of coming loose.

H. H. DELBERT.

Titusville, Penn.

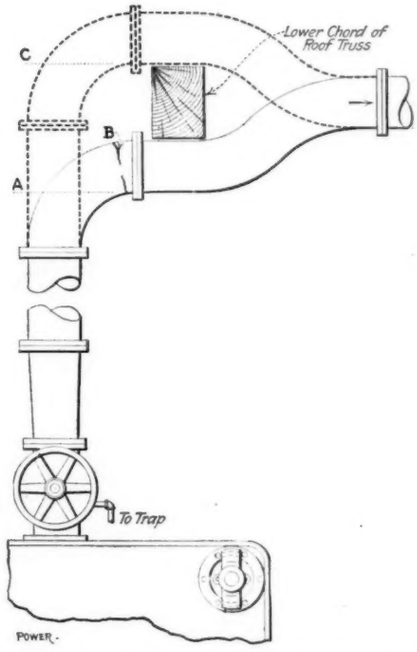
Damaged Steam Pipe Due to Faulty Design

At a cotton factory where I was employed the 10-in. pipe that conducted steam to the engine was arranged as shown by the full lines in the illustration.

The intention was to run the pipe in a straight line, but owing to a mistake, the lower chord of the truss that supported the roof of the engine house was so placed as to prevent it.

A special offset section of pipe was therefore made to go under the truss and the vertical piece shortened accordingly. In the elbow at the bottom of the vertical pipe is a ½-in. drip pipe, connected to a trap that is supposed to carry off the condensation accumulating in the pipe when the engine is not running.

Occasionally the drip arrangement fails to keep the pipe free of water, and if it is not noticed by the watchman, the result is a severe water hammer after the water gets above the line *A*. The offset section of pipe forms an ideal water pocket which, if permitted to go on filling, is sure to do damage. On one occasion the elbow was cracked at *B* and displaced by a new one, causing considerable expense and attendant interruption of service. Before the new elbow



PROPOSED CHANGE IN PIPING TO PREVENT ACCUMULATION OF WATER

was put in, I suggested turning the offset piece of pipe up, to go over the truss beam and lengthening the vertical piece to meet it, as shown by the dotted lines.

I explained that if the trap failed to work, no harm would result while the vertical section is filling with water, because the action is slow and the steam cannot mix with the water, therefore no water hammer could occur, and after the water reached the level of line *C*, it would trickle back to the boilers.

My suggestion was not approved by the company's representative, who said the pipe was designed by an expert and was all right. This answer was about as sensible as the mother's when a doctor asked about the milk she was feeding the sick child, and she replied: "Oh, I guess the milk is all right, because it says 'Fresh Country Milk' on the milk wagon."

Later on, either the trap failed to work or the valve in the connecting pipe may have been shut, for the 10-in. pipe broke and the steam rushed into the engine room until the boiler stop valves were closed. The cost of this break must have been considerable, because aside from the expense of replacing the pipe, the steam spoiled three large leather belts, and the mill was shut down until new belts arrived and were adjusted to place.

After this experience one would suppose that the company would surmise that something was wrong and not take further chances. The fact is, the pipe was replaced just as originally installed, which is evidence of their confidence in the judgment of their expert.

Under such management, when the cost of maintenance and operation are figured, they must indeed be suffi iently high to give the solicitors for the central stations and hydro-electric transmission companies an opportunity to put up a good argument in favor of stopping the local plant and buying power from the companies they represent.

J. W. PARKER.

Clinton, Mass.

Stopping Leaky Rubber Diaphragms

Frequently, after rubber diaghragms have been used for some time, they become slightly porous and give considerable trouble in the form of mysterious little leaks. This is especially noticeable in heating systems, controlled by electric thermostats and diaphragms operated by low-pressure air.

Recently I learned of a cheap and effective way of overcoming this difficulty. Whenever it became difficult to maintain the air pressure by a small hydraulically operated air compressor, a helper was sent out to try the rubber diaphragms with a brush and soapy water to detect leaks. If any were found, as evidenced by bubbles forming over the surface, the diaphragms were removed and the inside

surface given a good coat of ordinary molasses, after which they were replaced. Many of these treated diaphragms gave longer and better service than new ones put in at the same time.

It is now this concern's practice to treat all of its leaky diaphragms with molasses and mark them with the date to keep a record of the service they give. It has been found that diaphragms made of canvas indurated with rubber and treated with molasses when signs of leaks begin to show, are the cheapest and most satisfactory.

The idea was borrowed from a similar treatment of old porous bicycle tires and found to be even more satisfactory. The molasses seems to have no detrimental effect on the rubber, but rather tends to keep it pliable without rotting it.

M. S. WEIL.

Philadelphia, Penn.

Firebrick Walls Prevent Smoke

We have had much trouble with the smoke inspector while using South Fork scalp level bituminous coal of the following properties in a 250-hp. water-tube boiler with 0.2-in. draft in the uptake.

	Per Cent.
Fixed carbon	72.56
Volatile matter	15.39
Sulphur	0.9970
Ash	9.15
Moisture	2.09
B.t.u. commercial	13,825

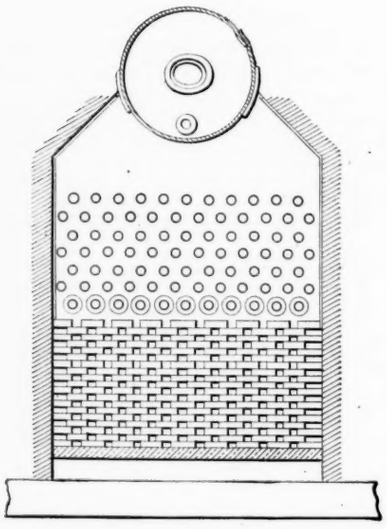


FIG. 1. END SECTION SHOWING BRICK SETTING

After obtaining prices on various devices for lessening smoke, we concluded that it would be cheaper to install more hot surface of such character that it might be easily repaired, and, if possible, of cheap material. We planned the surfaces shown in Fig. 2, using No. 1 firebrick at $20 per 1000. Four walls, *A B C D*, were built up dry to within 2 to 4 ft. of the tile under the tubes, and as shown in Fig. 1, the bricks were laid with spaces between making a checkerwork. In this way we added 84,825 sq.in. of surface to store the heat of the flames, which considerably lessened the amount of smoke.

FRANK MARTIN.

College Point, N. Y.

Taking Care of Drips

In manufacturing plants, when the area of the building is not over 100 sq.ft. and is run up a few stories, taking care of the drips is not simple, but in a large manufactory where the buildings are scattered over three or more acres, the drip problem becomes very much more complicated. I have found, during several years' experience, that engineers generally do not pay the attention to drips that their importance would seem to require. Often the sewer seems to be the favorite way to get rid of them, and as long as they are out of sight they are out of mind. Unless they blow back, no further attention is paid to them. Then usually further additions are made to the plant and again the drips go to the sewer, until in a few years the amount of condensation running to the sewer becomes considerable. Drips can generally be taken care of properly if a little thought is given them.

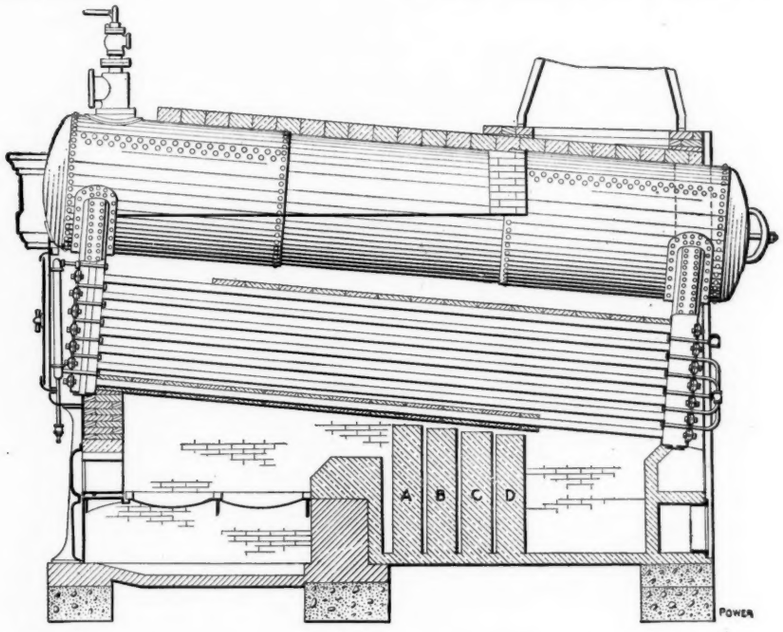


FIG. 2. SHOWING HOT WALLS FOR PREVENTING SMOKE

Plants occasionally grow beyond all expectations and, while the steam and exhaust lines will be enlarged, the same drip main is left to take care of the additional load. Sometimes it does, but often at the expense of half flooding the heating and drying apparatus, thereby not giving the results intended and possibly holding back the production, or putting more back-pressure on the engines. In such cases, a vacuum pump on the end of the lines will do the work nicely and will give quick results in circulation, and better drying, and the drip lines will be less likely to leak.

At some places the drips are not collected at all and cold water is fed through the heater. There generally is some waste in large plants, regardless of efforts to prevent it. Some managements go to the other extreme and spend more to save a few pounds of condensation than they will ever gain. Where the condensation amounts to a dozen or so pounds an hour, I have known long lines of drips to be run both over and underground, the cost and upkeep of which was out of all proportion to the saving effected.

Where the condensation eats the pipes rapidly it is occasionally cheaper to run a drip to the sewer; but it does not pay to forget about it. An inspection of traps at regular intervals is profitable in plants, large or small, for they always need looking after. I have seen traps that were in active service for about 15 years discharging into a river. Whether they were ever overhauled in all that time I could not find out, but when I saw them, they were certainly not holding back any steam.

If the condensation can be returned to the boiler that is the best disposition of it, unless there is considerable use for hot water when drips may be so utilized. Dripping the high-pressure into the exhaust is one way of saving long runs of pipe, but this cannot be done where much low-pressure steam is required, as

considerable of the discharge from the high-pressure main will not reëvaporate. I have found that a well in the exhaust, where a nest of high-pressure traps discharge, having a large low-pressure trap, answers the purpose very nicely.

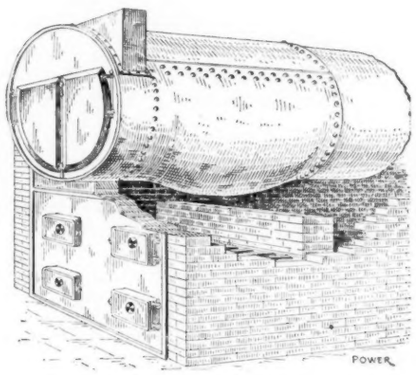
It is false economy to use an old trap on work for which it is not intended, rather than buy a proper one. How to care for drips is usually an individual problem, as conditions vary in different plants, and what would be suitable in one would not do in another. Hence no hard-and-fast rules can be given. The main trouble is that the drips are often neglected, whereas they should receive careful attention if the plant is to be run economically.

JAMES ROONEY.

Providence, R. I.

Badly Bagged Boiler

Herewith is shown a bagged boiler sheet extraordinary for the size and cause of the bag. I was not engineer at the plant when the boiler was burned, Feb. 12, 1912, but I took charge Feb. 27.



A 60x42x9 IN. BAG IN A BOILER PLATE

It is a return-tubular boiler 60 in. in diameter and 16 ft. long, and has fifty-six 3½-in. tubes. The bottom is made up of two 8-ft. sheets extending half way up on each side, having two double-riveted longitudinal seams and one girth seam. The bag was in the half sheet directly over the grates. It commenced about 3 in. from the girth seam, extended toward the front end of the boiler for 60 in., was 42 in. wide and dropped down 9 in., as illustrated.

No scale was in the boiler, but the plate was covered with about ⅛ in. of baked cylinder oil, due to letting the water get low in the heater (an old return-tubular boiler used as an open heater), so that cylinder oil found its way into the boiler. As near as I can find out, the boiler always had plenty of water.

On Monday morning, Feb. 12, after the engineer had started the engine and gone back to the boiler to throw a fire, he discovered the bag. The gage glass was nearly full of water and the pressure was 60 lb. No other reason being apparent, the oil must have caused the bag. The plant has two boilers, both the same size, but only one is used at a time.

The other boiler was in operation and it made me rather uneasy to know that I was firing a boiler undoubtedly coated with oil as the other had been. I had no chance to clean it until the other was ready for use, but I managed to keep the heater full of water and let some overflow to remove the oil from the surface of the water, and prevent more oil getting into the boiler. I commenced using soda ash, using two quarts a day for the first ten days, then feeding it every other day for the next ten days, and twice a week after that. By frequently blowing down and washing out the boilers, I have got rid of all the oil, and the boilers are now in good condition.

This shows what oil will do to a boiler, and also suggests a remedy for anyone that may happen to experience the same trouble.

WILLIAM DEBO.

Mansfield, Ohio.

Blowing Safety Valves Waste Steam

An employer once said to his engineer, when remonstrating about an unexpected discharge of the safety valve: "It was the most terrible, unearthly noise I ever heard, and if it occurs again you may consider yourself discharged."

"Well," replied the engineer, "there's this much about it; if you hadn't heard that noise you might 'a'heard a dern sight bigger one."

These expressions are perhaps typical of the general ideas regarding the purpose of the safety valve: a necessary nuisance, from the owner's standpoint, because custom, or the inspector, compels attaching it to the boiler. The engineer knows that it is to act as a relief for an unexpected or unavoidable increase of pressure beyond the safe limit of the boiler.

In many plants the pressure carried is but little lower than the pressure allowed, and a slight decrease in the demand for steam will cause the safety valve to open; if no objection is made this may be repeated at short intervals while the plant is being operated. It may be that the engineer is demonstrating his skill as a fireman, foolishly thinking that each blast proves his ability to master the steam-generating equipment, or, possibly the working pressure is carried as near the blowing-off point as possible to effect a short cutoff on the engine, for the sake of economy. The cost of the fuel wasted by the safety valve will generally offset the gain that may be credited to the shorter cutoff produced by the few pounds added to the initial pressure.

By an actual test on a boiler blowing off steam for four consecutive minutes, it was demonstrated that 6 cu.ft. of water, or 374 lb., was converted into steam and wasted. Taking 8 lb. of water as converted into steam per pound of coal burned, the amount of coal wasted in the four minutes was 46.7 lb.

CHARLES F. ADAMS.

Lynn, Mass.

Feed Pump Troubles and Repairs

When our feed pumps were run fast, one side would short stroke, the feed line shake and the check valves on the feed line slam continually.

I first examined the discharge pipe inside the horizontal tubular boilers and found it broken off in No. 2 boiler 2 in. from the reinforcing bushing inside the shell. This caused the slamming of the check valves as the feed water entered the steam space at 208 deg., while the temperature of steam at 120 lb. is 350 deg., therefore, causing a partial water-hammer. I then ground the check valves to a good seat on each boiler.

I next examined the water end of the pumps and found the plungers badly scored, and the suction and discharge chambers contained scale and fine pieces of concrete which caused the scored plungers. The pumps got their water from a tank which received the returns from the heating systems in 14 buildings.

The return tank was 12x10x8 ft., built of brick and lined with concrete ½ in. thick. The continual expansion and contraction eventually cracked the concrete and pieces of it then worked their way to the pump as the suction pipe had no strainer and the suction was taken from the bottom of the tank. I changed this so that the suction would be about 6 in. from the bottom and put on a strainer. The suction valves were in fair condition, but the discharge valves were too soft and the pressure had pushed them down between the spider arms of the valve seat. I ordered composition valves of the same material that is used in making disks for globe valves. I then had the valves faced and a groove made in them where they rest over the arms of the seat.

My reason for facing them was that after the valves are in use a short time the arms of the valve seats wear a ridge in the valve, and when it is turned by the continual opening and closing, this ridge keeps the discharge valves from closing or seating perfectly. Two new plungers finished the job and we had two good running pumps.

JOHN J. MAHONEY.

North Cambridge, Mass.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Water in the Low Pressure Cylinder

I have read the article "Water in the Low Pressure Cylinder" on page 898 of your issue of June 25.

I do not believe that this can be the explanation of damage from water in the low-pressure cylinder, because, the admission being only 0.3 of the stroke, the water cannot enter except during this period, but it has at least 0.8 of the stroke to get out of the cylinder. How, therefore, can it stay there to make trouble on the return stroke?

The case cited could produce damage to the high-pressure cylinder, because the water in the receiver could be drawn into this cylinder just at the moment when the exhaust valve closes for compression, when the engine was operating with so small a load that the pressure in the high-pressure cylinder fell below the pressure in the receiver.

H. BOLLINCKX.

Brussels, Belgium.

Improving a Neglected Plant

Mr. King's letter under this heading in the May 21 issue appeals to me as I have succeeded after much effort and with comparatively little expense in improving my plant, which is a central station.

When I took charge I found a most incompetent management. The manager acted as chief engineer with a first assistant who was too inexperienced for his responsibility. My coming was not pleasant to them as my contract called for a salary unusually high in that locality, and entire supervision of the plant. I was also to receive a 10 per cent. increase in salary every year until a stipulated amount had been reached. Should I fail to produce good results I was to resign. In six months my salary was increased 20 per cent.

I soon gained the support of the crew, as I had insisted on their highest efficiency and promised additional wages to each man in proportion to his worth. The men were instructed that those not meeting the requirements would have to go.

After putting the engines in good condition, I installed oil burners in the boiler furnaces, thus doing away with coal, and the service of two men. The wages of those retained were then raised. The feed-water heater only heated the water to 90 deg. and it was changed so that the water was heated to 210 deg. without using live steam. There were no feed pumps, injectors being the only means of feed, but I ordered pumps the first day I took charge. The oil-soaked wooden floor of the engine room was replaced by one of concrete. The swampy ground around the plant was filled in, and the drainage pipes repaired. A new foundation was also built under one engine.

The service has so improved that our output is greatly increasing and new machinery is being installed to meet the demands. Two large generating sets have recently been added and an order has been placed for a large Diesel oil engine.

My experiences teach me that if the management will coöperate fully with its engineer, he will furnish the highest kind of service at a minimum cost. Most chief engineers will amply reward their assistants if they prove competent and progressive.

JOHN PIERCE.

Chicago, Ill.

Emergency Expansion Bolt

The expansion bolt mentioned by Mr. Daly in the May 28 issue would answer its purpose admirably, but most expansion bolts when once put into a wall

SLEEVE FOR EXPANSION BOLT

cannot be removed without cutting them out, thus defacing the wall. For this reason the ordinary expansion bolt is losing favor and going out of use while one that can be removed and will allow the fixtures or machines to rest against the wall without having to be sprung out over the ends of the bolts is most desirable.

I put a bracket back of a large pipe, which extended about 6 in. from the wall, to support the pipe and also to keep it from vibrating. The bracket, which was made U-shape, could not be sprung out over the ends of bolts, and other means had to be resorted to. As the wall was very thick, and long bolts could not be passed through, some scrap lead was melted into sleeves 3/8x7/8x4 in., as shown, and sawed down from each end in quarters for about 1¼ in., leaving about 1½ in. of solid tube in the center.

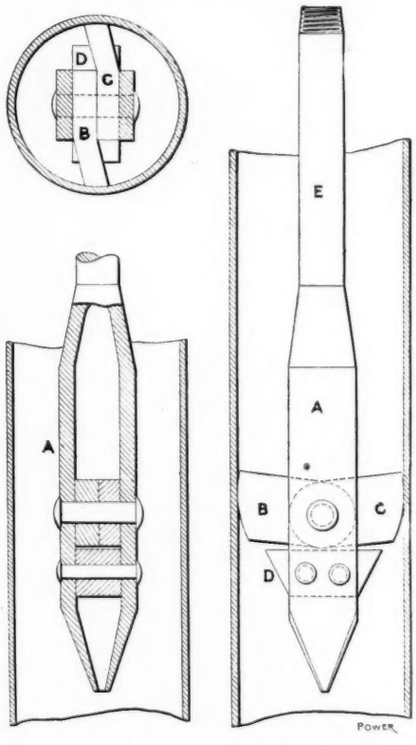
These were driven in the holes in the brickwork flush with the face of the wall and ½-in. lag bolts were then screwed into these lead sleeves, expanding them in the brickwork and making a very rigid, quick and neat job. The taper of the lag-bolt thread is sufficient to expand the lead sleeve and the coarse wood-screw thread gives it a very firm hold. Care must be taken not to have the center hole in the lead sleeve too small for the bolts or there is risk of breaking off the bolt before it can be screwed in tight.

R. A. CULTRA.

Cambridge, Mass.

Lifting a Broken Plunger

F. H. Cordner's letter under this heading in the May 7 issue was particularly interesting to me as I have had much experience in boring wells and lifting broken plungers, tubing, pipes, etc., from them.



DEVICE FOR LITING DEEP WELL PLUNGERS

The tool illustrated is the one with which I have had the greatest success. It is made by welding together two pieces of 1½x3/8-in. iron *A*, shaped so as to re-

ceive the toggle members *B* and *C*. A square piece of iron *D* is riveted on, as shown, so as to act as a stop for the pieces *B* and *C*. By welding a short length of round iron *E* to the piece *A* and threading it to receive a length of pipe, additional weight is given the tool, which is desirable.

The tool is lowered into the well with a rope and the toggle pieces bind so hard that there is no slipping. The tool is very easily disengaged from the pipe or tubing.

W. B. ODELL.

Birmingham, Ala.

Putting in Crankpins

Mr. Gilson's method of putting in crankpins, as described in the June 11 issue, would be rather slow. By heating the hole with pieces of hot shafting, I do not think he could get the disk hot enough.

With a pin of the proper size and the disk sufficiently hot, the pin will go in without any driving, but one must work in a hurry, for once the pin has entered the hole it will soon heat and expand enough to become tight. To be on the safe side, a ram should be made so that the pin could be driven in if it stuck. I prefer pulling a pin in with a bolt. Before the pin is taken out of the lathe, bore and tap the end for the pulling bolt.

For a 5-in. or a 6-in. pin use about a 1¼-in. machine-steel bolt. When all is ready give the pin a coat of white lead, square the pin with the disk and pull up on the nut. Be sure the bolt is a snug fit in the pin; otherwise it may pull out. Lightly tap the end of the pin as the bolt is pulled up; usually swing a piece of shafting from over head, but if there is not room, a hammer may be used. After the pin is pulled up, saw off the pulling bolt. I have put in several pins this way and have never heard of any giving trouble.

P. P. MAJORBANKS.

Bedlich, La.

Mr. Gilson might have saved fully 50 per cent. of the drilling work if, instead of drilling the holes near the circumference of the pin, he had drilled on the diameter, as shown in the illustration. As the ratio of one-half the circumference to the diameter is 1.5708 to 1, the effect in loosening the pin would have been the same.

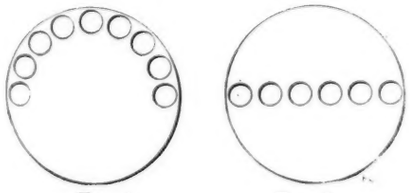


FIG. 1 FIG. 2

POSITION OF HOLES FOR REMOVAL OF PIN

Fig. 1 shows nine ⅞-in. holes on a 6-in. pin which are necessary, according to Mr. Gilson's way of removing the pin. In Fig. 2 is shown six ⅞-in. holes on the same size pin, which are enough to allow it to be readily removed.

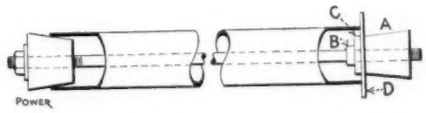
CHARLES F. PRESCOTT.

Los Angeles, Calif.

Unnecessary Risks

In commenting on the editorial in the Apr. 30 issue, Charles J. Mason, under the above heading, in the June 4 issue, writes of a risk which he deems necessary.

I have plugged the tubes of Scotch marine boilers, but never deemed it necessary to toast myself over a bed of hot coals to do so. We used a cast-iron plug *A*, as illustrated, instead of a pine one, and the rod, plug, washer, etc., were pushed through the tube from the front end. The greatest diameter of the plug was slightly less than that of the tube. A shoulder *B* was made on the rod at the end of the threads, and the washer *C* was put on over the threads and against this shoulder. A piece of asbestos packing *D*, the diameter of which should be larger than that of the tube, was then slipped on over the rod and against the washer. The threaded plug was then screwed tight on the rod and the whole thing pushed through to the rear end of the tube with the aid of an extension handle.



PLUGS IN BOILER TUBE READY FOR TIGHTENING

The rear plug should be pushed a little beyond the end of the tube to allow the asbestos packing to straighten out. The rod should then be pulled hard forward and tightened with the nut at the front end. The purpose of the packing is to enlarge the diameter of the plug and seal the joint.

LEIGHTON JOHNSON.

Exeter, N. H.

How the Cylinder is Cut

The cutting of the top of the cylinder bore, described by William E. Dixon, in the June 11 issue, brings to mind a similar experience I had with a small engine. The first indication of trouble was a pound in the cylinder when running at about one-fourth rated load, which continually increased. When the cylinder head was removed it was found that the top of the bore was badly cut and the piston rings were very rough.

A good grade of oil was used in the plant, and other engines using it gave no trouble. Nevertheless, the cut cylinder appeared not to be receiving proper lubrication. Steam at 105 lb. pressure without superheat was used. I believe that even some of the high-grade oils will not mix with steam at this pressure, but will travel along the steam pipe and steam chest in drops as it is fed, and upon entering the cylinder very little of it comes in contact with the top, which would account for the wear at that point.

After repairing the cylinder and fitting new piston rings, it was decided to feed oil directly into the top of the cylinder at the middle of its length. A ⅛-in. hole was drilled into the cylinder and enlarged and tapped for a ¼-in. pipe, on which was connected a lubricator. Although little oil reached the top of the cylinder, it stopped the wear on both rings and cylinder.

That the other engines were not affected seems to be due to a difference in the iron in the cylinders, which is much harder and the stroke is also shorter. Since then a cheaper grade of oil has been used, and it seems to mix with the steam and reaches all parts with which the steam comes in contact.

P. L. WERNER.

McKeesport, Penn.

All the symptoms described by Mr. Dixon tally closely with those I once experienced in a similar case, and the whole trouble was corrected by squaring off the ring, i.e., making the ring freer in the groove.

Some engineers are inclined to blame things that go wrong to almost any cause but the right one. For instance, superheated steam, boiler compounds, improper methods of lubrication or poor lubricating oil are often blamed, while sometimes the whole fault lies in lack of care to see that the piston rings can expand and contract freely before being placed in the cylinder. This very same cause, and its effect, that is, cutting a cylinder out at the top, once cost a well known engine builder a new cylinder.

Steam engineering is a profession from which all possible "ifs" and doubts should be eliminated. The surest way to settle any doubt about a ring being in running condition is to put the piston together complete, securing the follower as tightly as though it were in the cylinder, and then try a "feeler" all around the ring to make sure that it is somewhere near parallel. A light pressure, or a tap with a hammer handle, or block of wood at any point on the circumference of the ring, should move it to the side opposite. If it does not move freely, it should be

taken out, and the marks made by the contact of the surfaces will show where it should be scraped to clear the ring. It is well also to slightly round the edges with a file, as this removes all sharp corners and stops the tendency to scrape the lubricant from the cylinder surfaces.

One engine builder makes a piston with one ring, such as is described by Mr. Dixon. To prevent the ring from wearing a shoulder in the cylinder, a groove is cut at each end of the bore for the edge of the ring to come into, when the piston is at the end of each stroke.

H. R. Low.

Moosup, Conn.

The cause of the excessive ring wear is too much spring pressure. To overcome this, Mr. Dixon should straighten out the springs until they are all in place under the ring and the ring held flush with the face of the piston at a point opposite the cut. He can force the two ends together with a strong thumb pressure. If the springs will not straighten out by hitting them with a hammer, heat them to a cherry red and quench in ordinary engine oil. To get a uniform curve in each spring, lay one on the floor and mark its curvature with a soap-stone pencil, then heat the spring and straighten it to suit conditions.

The reason all the cylinders are not affected the same is due to several causes: variation in the hardness and set of the springs, also difference in the hardness and density of the metal in the rings. Very often it is due to the manner of casting the cylinders. Some builders cast all cylinders vertically with a 6- or 8-in. riser all around the cylinder bore. This allows sand, cinder and also the plumbago wash that becomes detached with other impurities to float to the top. This is then cut off in the shop and thrown away. Invariably such cylinders have metal of a uniform hardness and density in the bore. Others cast their cylinders horizontally, with the result that the impurities are occluded in the iron, either in the top of the cylinder, if cast with the steam side up, or on the bottom, if cast exhaust side up. Usually cylinders made this way have a soft, spongy spot somewhere in the bore.

Rings with too much spring pressure on them soon wear to a sharp edge that will act like a planer tool and cut even the hardest iron. The top of the cylinder bore gets little lubrication, unless the oil is thoroughly vaporized, or special provision is made to put it there, as in gas-engine practice. I would suggest that Mr. Dixon fit the pistons tighter in the cylinder bore and apply minimum spring pressure behind the rings, and also put a special oil pipe in the top of the cylinder.

His particular conditions will readily suggest how best to do it. Until he gets over his trouble, he should take out the piston rings at regular intervals and file off the sharp edge. If he follows the above suggestions, he will save the cylinders that have not yet been affected.

J. F. Nagle.

Cos-Cob, Conn.

That Red Hot Blowoff Pipe

In the issue of June 18, Mr. Hannemann inquires why his blowoff pipe became red hot if it was full of water. The natural reply would be that it did not. If the pipe was full of water it did not become red hot; if it became red hot it was certainly clear of water at that point.

Most probably mortar, or other rubbish in the boiler, got into the blowoff pipe and obstructed it; this sometimes happens with boilers just erected.

So much for Mr. Hannemann's troubles. I will mention some of my own experiences which run along a similar line. I used to be "head fireman" in a plant equipped with vertical water-tube boilers, fired with underfeed stokers and using mechanical draft. We had as nearly perfect feed water as I ever saw, making practically no scale. In those days, generation of steam was my specialty, and I took pride in keeping the safety valves just hissing all day.

Sometimes the load would increase and only an incandescent fire would maintain the pressure. Imagine my astonishment to find that the fire side of the tubes would bulge out with little bags about as large as half an egg. I had always been taught by precept and common sense that a clean sheet or tube could not bag, but here was a contradiction, for the tubes were as free of scale as when they were put in, and no oil got in our boilers. The only explanation we could give was that the intense heat impinging against the tubes would generate steam, making a bubble inside the tube, holding the water away for an instant, allowing the tube to become red hot and the pressure to bulge it. I never knew of a burned blowoff pipe which was not obstructed.

Recently a horizontal boiler in a large sawmill plant, fired with dry shavings, had the rear head among the tubes burned without the water being low in the boiler. We accounted for that only by the possibility of steam bubbles inside and by the fires being pushed to the limit.

C. R. Summers.

Chattanooga, Tenn.

Mr. Hannemann's trouble is a fair sample of power-plant happenings. I do not deny that a blowoff pipe could get red hot, but my experience with men in plants leads me to believe that he entertained a false impression.

With a boiler subject to an intense fire, one is apt to imagine when looking at a soot-covered sheet over the fire, that the sheet is bagged. I know of an engineer of 30 years' experience who, on looking under his boiler, became excited and declared it was red hot, while two gages of water showed in the glass. He pulled the fire and sent for an inspector, only to be told that he must have imagined it.

Mr. Hannemann is burning shavings which would cause a flame to travel the entire length of the boiler. On opening the rear door and looking at the boiler under these circumstances, one is apt to think the blowoff pipe and everything else is red hot. This is especially probable since the boiler was being put into service for the first time, and on such occasions an excess of anxiety conduces to such delusions.

Mr. Hannemann inquires as to the best way of drawing a sawdust fire; I would not draw one even in an emergency. Closing the ashpit and opening the fire-doors, and then covering the fire with about 8 in. of sawdust is best. The dampers should be left open.

A. Waldron.

Revere, Mass.

Soon after a fire was built under Mr. Hannemann's boiler, heat bubbles began to gather and move out through the open end of this pipe into the boiler until, as the heat became more and more intense, the amount of steam generated in this pipe was sufficient to keep the water driven back out of the pipe. Consequently, it is very plain why the pipe became red hot. The powder substance mentioned by Mr. Hannemann, was probably lime scale.

A careful perusal of the underlying principles upon which the Solignac-Grille boiler is based, as described on page 867 of the June 18 issue, would help Mr. Hannemann to understand why his pipe became red hot, since what they are proposing to overcome is analagous to this blowoff-pipe problem.

Lloyd V. Beets.

Nashville, Tenn.

Flue Gas Analysis

Commenting on the editorial under the above heading in the June 11 issue, I would state that six months ago I knew practically nothing of flue-gas analysis. Our evaporation per pound of coal from and at 212 deg. F. was 5.5 lb. Now it is over 10 lb., almost 100 per cent. gain.

The engineer who does not give flue-gas analysis his attention may some day find himself at the end of a wheel-barrow.

George C. J. Spangenberg.

Buffalo, N. Y.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Connecting Indicator to Cylinder Drip Cocks

Can true indicator diagrams be obtained of an engine by connecting the indicator to the cylinder drip cocks?

E. W. H.

Yes, if the indicator is connected at right angles to the drip connection, or otherwise, so as to prevent impact of the drip water against the indicator piston, and there is thorough drainage of the connections, as by a drip tee placed in the engine side of the indicator to drain the connection and leave unobstructed the full area of steam passage required for proper action of the indicator piston. When all other conditions are constant under which a set of diagrams are taken, if the diagrams are the same, it should be evidence that they have been unimpaired by presence of water in the connections, though such evidence is not entirely conclusive.

Coal Saved by Warmer Feed Water

Where 2400 lb. of coal is used per day for making steam at 100 lb. pressure, which is used for a hoisting engine and for pumping, the temperature of the feed water being 45 deg. F., how much coal should be used for the same work with feed water at 100 deg. F.?

S. T.

A pound of water raised from 32 deg. F. to steam at 100 lb. boiler pressure requires 1188.8 British thermal units. With feed water at 45 deg. F. each pound requires 45 — 32 = 13 B.t.u. less than at 32 deg., i.e., each pound requires 1188.8 — 13 = 1175.8 B.t.u. With feed water at 100 deg. F. each pound would require 100 — 32 = 68 B.t.u. less than at 32 deg., i.e., each pound would require 1188.8 — 68 = 1120.8 B.t.u. Therefore, the amount of coal required with feed water at 100 deg. should be

$$\frac{1120.8}{1175.8} \times 2400 = 2287.736 \text{ lb.}$$

Capacity of Water Tank

The inside dimensions of a circular tank are 12 ft. diameter at the top, 15 ft. diameter at the bottom and 15 ft. deep. What is its capacity?

T. S.

The cubical content of a circular tank of the form described would be found in cubic feet, by adding together the area at the top, the area of the bottom, and four times the area of cross-section taken at the mid-depth; then multiply the sum by the whole depth and take 1/6 of the product. In the example the areas to be added would be:

Area at top = area of 12 ft. diameter = 113.10 sq.ft.

Area at bottom = area of 15 ft. diameter = 176.71 sq.ft.

The mid-section being 13½ ft. diameter four times its area would be 4 × 143.14 = 572.56 sq.ft. and their sum = 862.37 sq.ft.

Then 862.37 × the depth, 15 ÷ 6 = 2155.925 cu.ft., equivalent to 2155.925 × 7.48 = 16,126.319 gal.

Slip of Water Meter

Why should the registration of a water meter in the feed line to a boiler be very much less for a given quantity of water fed through it by a double acting, single-cylinder 4½x2½x6-in. pump than when fed through a meter by a 6x4x6-in. duplex pump?

H. H. B.

Intermittent discharge from the single-cylinder pump would cause most forms of meters to register less than when the same quantity is discharged at a steadier rate of flow as would be the case when feeding the boiler with the duplex pump.

An additional cause for greater slippage of the meter when using the smaller pump may be that it is used at a rate of discharge which is very much lower than the rate of discharge of the duplex pump is lower than the rated capacity of the meter.

Chimney Size and Construction

What should be the size of a chimney suitable for 300-hp. boiler plant using coal as fuel with natural draft? Would vitrified brick be a good material for such a chimney?

T. W.

If the chimney is built 85 ft. high it should have a round flue 64 in. inside diameter or if square 44in.x44in. at the smallest part. Vitrified brick would be good for the outer shell if a mortar is used which will set up hard and with good adhesion to the brick; but the interior of the shell should be protected from heat by an independent core wall of fire brick or ordinary unvitrified brick. The protecting core should be built not less than 8 in. thick for a height of 10 ft., and 4 in. thick an additional 10 ft., and so constructed that it may expand independently of the main wall. For this purpose about ½ in. space should be left for lateral expansion of the core in every direction and there should be plenty of room for independent vertical expansion.

Hoisting Engine Link Motion

Why is it that, when the link is down or in the forward motion, the valve of a hoisting engine should travel farther than when the link is up, or in the backward motion.

A. C. Z.

An ordinary link motion valve gear cannot be so designed that the same travel will take place for forward as for backward motion without sacrificing the equilization of cutoff.

Using Averaging Planimeter

Taking an indicator diagram from an engine with the reducing motion so adjusted that the length of the diagram is 4 in. and appiying a direct-reading planimeter, it is found that the engine develops 250-hp. Then changing the reducing motion, but with all other conditions the same as when the first diagram was taken, another diagram is taken and upon applying the planimeter, still adjusted for a diagram 4 in. long, a lower figure than the first is obtained owing to the fact that the second diagram has smaller area than the first. How are the calculations to be made correctly?

K. W. K.

By using a planimeter which measures the area, the mean effective pressure is obtained by dividing the area by the length and multiply by the scale of the spring, i. e.

$$\frac{Area}{Length} \times scale = m.e.p.$$

For the same scale the same m.e.p. could not be obtained, unless the value of the fraction remained constant. If the diagram is 2½ in. long in place of 4 in. long the area would be reduced in the same proportion so that if the adjustment of planimeter is employed which is proper for 4 in. diagrams, multiply the reading by 4 and divide by the actual length of diagram.

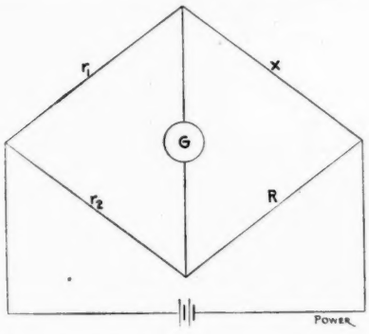
If the planimeter used gives direct readings for different lengths of diagrams then with a shorter arm set for a shorter diagram, the wheel movement will be proportionately greater for the same area.

Study Questions

This Week's Questions
Last Week's Answers

(56) A man painting a steel stack is sitting on a sling seat hung by a single rop running through a pulley at the top of the stack. The free end of the rope is fastened to a ring in the ground 54 ft. from the base of the stack. When the man is 62 ft. from the ground the length of rope from the ring through the pulley to his seat is 100 ft. How high is the the stack?

(57) In the accompanying diagram of the connections for a measurement of resistance by the Wheatstone bridge, the



resistance r_1 is 10 ohms, r_2 100 ohms, and when no current flows through the galvanometer G, the resistance R is 343 ohms. What is the unknown resistance x?

(58) If a steam engine had thermal and mechanical efficiencies of 100 per cent., how much heat would it consume in raising a load of 1 ton to a height of 500 ft.?

(59) A beam 15 ft. long, supported at both ends, has a uniform load of 5000 lb., a concentrated load of 3000 lb. 5 ft. from one end and another of 4000 lb. 5 ft. from the other end. What are the weights on the two supports?

(60) Shot vertically with a velocity of 1000 ft. per second, what height will a cannon ball reach, neglecting air resistance?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(51) The work done per minute is 15,000 × 200 = 3,000,000 ft.-lb. corresponding to

$$\frac{3{,}000{,}000}{33{,}000} = 90.909\ hp.$$

Dividing by the mechanical efficiency gives

$$\frac{90.909}{0.70} = 129.87\ i.hp.$$

In the formula

$$I.hp. = \frac{plan}{33.000}$$

or

$$p = \frac{33{,}000 \times i.hp.}{lan}$$

where

p = Mean effective pressure, pound;
l = Length of stroke, feet;
a = Area of piston, square inches;

l is given as 3 ft.; n is twice the 60 r.p.m., or 120; p is to be determined and a the average area of the two sides of the piston can be found from the formula.

$$a = \pi\left(\frac{D^2}{4} - \frac{d^2}{8}\right)$$

where

$\pi = 3.1416$;
D = Diameter of piston;
d = Diameter of piston rod.

Hence

$$a = 3.1416\left(\frac{20^2}{4} - \frac{3.5^2}{8}\right) = 309.35\ sq.in.$$

Then

$$p = \frac{33{,}000 \times 129.87}{3 \times 309.35 \times 120} = 38.48\ lb.$$

(52) The consumption per hour will be

6000 × 1.75 = 10,500 lb.

and for 24 hr.

10,500 × 24 = 252,000 lb.

or

$$\frac{252{,}000}{2000} = 126\ short\ tons$$

or

$$\frac{252{,}000}{2240} = 112.5\ long\ tons$$

(53) The current flowing, by Ohms' law, is equal to the electromotive force divided by the resistance

$$I = \frac{E}{R}$$

As it is the same throughout the circuit

$$I = \frac{50}{300} = \frac{35}{x}$$

$$x = \frac{35 \times 300}{50} = 210\ ohms$$

(54) The volume of a sphere is

$\frac{1}{6}\ \pi D^3$ or $0.5236\ D^3$

where

$\pi = 3.1416$;
D = Diameter of sphere.

Then the volume of the 2-ft. sphere is

$0.5236 \times 2^3 = 4.1888$ cu. ft.

This sphere with 1 cu.ft. added uniformly will become a sphere of a volume of

4.1888 + 1 = 5.1888 cu.ft.

and

$$5.1888 = 0.5236 \times D^3$$

$D^3 = 9.9099$;
$D = 2.148$;

is the diameter of the new sphere. Half the difference of the diameters will be the thickness of the covering or

$$\frac{2.148 - 2}{2} = 0.074\ ft. = 0.888\ in.$$

or closely ⅞ in.

(55) The pressure corresponding to a head of water of 185 ft. is

185 × 0.433 = 80.105 lb.

On a diameter of 20 ft., or 240 in., in 1 in. length of cylinder, the pressure tending to rupture will be

$$\frac{240 \times 80.105}{2} = 9612.6\ lb.$$

This divided by the allowable unit stress gives

$$\frac{9612.6}{16{,}000} = 0.6008\ sq.in.$$

as the required section per inch of length, therefore the thickness for no joints, and for joints of 75 per cent. efficiency

$$\frac{0.6008}{0.75} = 0.801\ in.$$

is the required thickness.

Dallas Plant Enlarged

Dallas, Tex., is rejoicing over the success of its electric power plant. In the past three years the population has grown so rapidly that $600,000 have been spent in enlarging the plant to meet the increased demands.

Chief Engineer J. J. Markham, formerly of Pawtucket, R. I., has practically rebuilt the plant during his administration and has brought it to a high efficiency.

The plant now has 19 boilers, mostly of the water-tube, oil-burning type, having a total of about 8000 hp. General Electric Curtis turbine generating units ars used. The 7500-hp. turbine set recently installed is the largest in Texas. Probably more additions will be required in the next two years.

Loss by Pittsburgh Flood

On July 21, states a press dispatch, a terrific rain storm flooded Pittsburgh manufacturing plants, impaired the trolleys and railroad service and inundated many business houses. The plants of the Westinghouse Air Brake Co., Wilmerding, and the Westinghouse Electric & Manufacturing Co., East Pittsburgh, were flooded by several feet of water. The Westinghouse interests are said to estimate their loss at $175,000.

Electric Plant Purchase Approved

Judge Hollister, of the United States District Court, recently approved the sale by the trustees of the Hamilton Gas & Electric Co., of Hamilton, Ohio, to Frank L. Perin, who represents the majority of the bond holders, for $430,000.

Annual Convention of Ohio Electric Light Association

Cedar Point, one of Ohio's famous watering places, was the scene of the eighteenth annual convention of the Ohio Electric Light Association on July 16 to 19. The attendance was the best ever reaching almost 500, and with only one professional session daily, all had ample opportunity to enjoy the entertainment provided by the committee, the bathing or the attractions at the "point."

The sessions were held on the second floor of the convention hall and a number of interesting exhibits occupied practically all of the first floor. The papers were of practical interest, and a talk on "Rate Making" by Hon. Halford Erickson, chairman Railroad Commission of Wisconsin, was especially appreciated.

Tuesday afternoon at 2 o'clock the first session opened with the usual routine business, including the secretary's report covering the progress made by the society, financial conditions and some recommendations on matters of current interest.

President W. C. Anderson, of the Canton Electric Co., in his address referred to the work of the association and highly complimented the secretary on his efficient and energetic services. The president spoke of the improvements made in the Mazda lamp, increasing the efficiency, and of the new sizes as small as 15 watts with early prospects for a 10-watt lamp. The introduction of these lamps reduces the amount of current used and makes the question of minimum charge for house service of the greatest importance.

Fresh air and the danger of poisoning from carbon monoxide when using natural gas for lighting in poorly ventilated rooms was a topic touched upon with the conclusion that electrical energy was the only illuminant which did not contaminate or impoverish the air. After referring to the process of sterilizing water by means of electricity and the advancement made in electrical heating devices, the president closed with a few remarks on municipal vs. private ownership. In his opinion, the inefficiency and waste of municipal control far exceeded the profit obtained under private ownership.

Committee Reports and Papers

The material presented by the committee on electrical transmission was of a general character, dealing principally with construction data on local transmission systems and lightning protection. Figures on maximum and minimum spacing of conductors for various voltages were given and a good general rule on transmission of 1000 volts per mile of line suggested. The discussion centered on lightning protection by means of multigap and electrolytic arresters and an overhead ground wire. This material will be presented in a later issue.

On Wednesday morning, the Hon. Halford Erickson, chairman of the Railroad Commission of Wisconsin, told why rate regulation was necessary and pointed out the duties of the commission in regulating rates for electric-lighting plants. The rates must be based on a fair valuation of the plant. The cost of reproduction new, less depreciation, rather than original values is now favored by the courts. Mr. Erickson told how the original cost and cost of reproduction were computed and also gave considerable time to depreciation and its determination. In making rates, he considered the demand factor of the greatest importance. There was a big difference between the cost of production for a customer taking energy only one hour per day and another drawing on the supply for 10 or perhaps the full 24 hours. A fair profit over operating expenses and depreciation should be permitted. In other words, the rates should be such as would allow a free flow of capital and executive ability into the field. For electric lighting 8 per cent. on a fair cost value should be a reasonable return on the investment. An animated discussion followed and a great many questions were asked about the relative cost of serving one customer with 1000 kw. and 1000 customers with 1 kw. each, on raising and lowering rates by the commission and on discrimination in rates.

On the subject last mentioned O. P. Gothlin, chairman of the Ohio commission, emphasized the importance of putting honest values into the property. The promoter has no right to give away stock or sell it at nominal values and make the company pay dividends on the "water." In estimating the worth of a property the going value as well as the actual should be taken into account. A great deal of time, ability and money are required to work up the business to a profitable basis.

Secretary D. L. Gaskill followed with a paper on "The Public vs. the Utility," which was full of common sense on the proper relations between the public and the electric-lighting companies. For low rates and good service, protection from unwise and sometimes malicious competition must be assured by the state. The companies will then be able to command the capital necessary and the public-service commission can control and adjust any point of dispute on service or rates.

The committee on Costs, J. D. Lyon, chairman, submitted a report closely related to the two previous papers. All items entering into the valuation of a public utility were enumerated and some little space given to the following subjects: going value, depreciation and diversity factor. The money spent in building up the business and securing new customers should be included in the going value, and conditions determine whether these expenditures should be charged to construction, operating expenses or to cost of plant. The committee urged the importance of securing reliable data on the diversity factor and its effect upon the individual cost of serving. According to the character of the service required or perhaps the time of serving each customer should pay a certain proportion of the fixed charges. What this proportion shall be in each case or in each class of service is the question.

A committee was appointed to investigate the desirability of the society affiliating with the National Electric Light Association. Ohio has the largest state engineering association in the country so that a favorable report by the committee should prove of mutual advantage.

Meter accuracy and testing was the burden of the report by the meter committee. This was on Thursday morning and immediately following a paper on "Electrolytic Purification of Sewage," by Prof. F. C. Caldwell, closed the session.

"Electricity in Rural Districts" was the first paper at the Friday session. The author, J. C. Matthieu, spoke of the opportunities in this field and the problem of educating the farmer to the use of electrical energy for power and lighting. At present rural business does not pay unless the service is taken from a line installed between two urban centers.

In his paper on "Joint Pole Line Construction," J. L. Spore advanced some original ideas on the arrangement of wires on a joint pole line, and cited a particular joint-pole agreement covering all necessary restrictions and regulations governing expenses and maintenance. The discussion disclosed a joint construction-work agreement about to be made standard by the Bell company. An abstract will be included in the proceedings.

Entertainment

One professional session daily gave the visitors time to enjoy the bathing and not miss any of the features provided for their amusement. On Tuesday evening all were guests of the association at a banquet enlivened by popular songs between the courses. Dancing followed, and on the next evening another banquet at which "Walden," the magician, entertained. On Wednesday afternoon a water ball game between the supplymen and the engineers aroused a great deal of interest. The result was a defeat for the supplymen and the same thing happened in the ball game on Thursday. Musicales, card parties, a launch ride and bowling amused the ladies.

Officers Elected

For 1912-13 the following officers were elected: President, J. C. Martin, Wilmington; vice-president, J. D. Lyon, Cincinnati; secretary-treasurer, J. D. Gaskill, Greenville; executive committee, W. C. Anderson, Canton; O. H. Hutchings, Dayton; W. Parsons, Springfield; W. R. Griffen, East Liverpool and W. E. Richards, Toledo. The usual committees were also elected at this time.

Exhibits

Among the exhibitors were the usual companies making electrical products such as lamps, transformers, motors and cooking utensils, and in addition electrically operated washing machines were shown. The list of firms follows: Allis-Chalmers Co., Milwaukee, Wis.; The Brokaw Eden Mfg. Co., St. Louis, Mo.; The Canton Rubber Co., Canton, Ohio; Chicago Fuse Manufacturing Co., Chicago; The Copeman Electric Stove Co., Flint, Mich.; Crocker-Wheeler Co., Ampere, N. J.; The Eclipse Electric Manufacturing Co., Chicago; A. D. W. Electric Sign Co., Cleveland; The Enterprise Electric Co., Warren, Ohio; Fort Wayne Electric Works, Fort Wayne, Ind.; General Electric Co., Schenectady; The John Dietz Mfg. Co., Cincinnati; W. N. Matthews & Bro., St. Louis; Nelite Works of General Electric Co., Cleveland; The Ohio Blower Co., Cleveland; The Packard Electric Co., Warren, Ohio; The Premier Vacuum Cleaner Co., Cleveland; The Sterling Electric Mfg. Co., Warren, Ohio; The A. L. Sykes Mfg. Co., Cincinnati; The Triumph Electric Co., Cincinnati; Wagner Electrical Mfg. Co., St. Louis; Westinghouse Electric Manufacturing Co., Pittsburgh.

Graphite as a Surfacer

Lubrication is the very life of a moving mechanical part. If you neglect lubrication or do it on a hit-and-miss principle, you pay for it by broken parts, shutdowns and other kindred troubles. The road to the junk heap is made an easy one if insufficient or inferior lubricants are used.

No matter how well you may think a piece of machinery is lubricated, unless provision is made for the time when oil or grease fail to do their duty (caused by an obstructed oil channel, overload, leaks, etc.), very serious abrasion is likely to take place and it is then the lubricant which can stand up under adverse conditions, takes its knocks and is not thin skinned, that protects you against losses.

Prof. Goss strikes the keynote when he says, "Tests show that with no other lubrication than graphite the journal and brass may be run together for hours at a time, heating up to a high temperature, and this may be done repeatedly, and yet the rubbing surfaces of the bearings remain unscored. As a protection against abrasion of rubbing surfaces, I know of nothing that can equal the record of graphite."

Boiler Explosions in 1911

A summary of boiler explosions for 1911 in the United States has been carefully compiled by *The Locomotive*. The figures follow:

SUMMARY OF BOILER EXPLOSIONS FOR 1911

Month	Number of Explosions	Persons Killed	Persons Injured	Total of Killed and Injured
Jan.	76	22	43	65
Feb.	44	31	52	83
March	39	27	38	65
April	39	14	22	36
May	43	18	39	57
June	31	30	17	47
July	42	18	34	52
August	32	13	29	42
Sept.	29	7	20	27
Oct.	48	14	53	67
Nov.	39	11	35	46
Dec.	37	17	34	51
Totals...	499	222	416	638

Smoke Problem Lectures

The department of industrial research, University of Pittsburgh, was recently provided with a fund for thoroughly investigating the smoke nuisance. The investigation is being conducted by 25 specialists, seven giving it their entire attention.

Recognizing the interest in the smoke problem manifested by many American cities, the department announces that its staff is prepared to lecture on the following subjects: The smoke nuisance (a general presentation of its main phases); smoke and the public health; smoke and the cost of living; smoke and plant life; methods and means of smoke abatement; the effect of smoke on buildings and building materials; the psychology of smoke; the smoke nuisance and the housekeeper. For further particulars apply to Dr. R. C. Benner, Department of Industrial Research, University of Pittsburgh, Pittsburgh, Penn.

Alternating Current Generators

Bruce Peebles & Co., Ltd., engineers of Edinburgh, have recently issued pamphlet 13B, which gives very complete ratings of the European standard 50- and 25-cycle machines for a range of speeds of from 1000 to 125 r.p.m. at pressures from 200 to 6600 volts. Rules are given for selecting the correct machine for other cycles not listed. The pamphlet also contains full lists of approximate weights and dimensions of alternators with exciters. The Peebles machines are manufactured to meet all special requirements.

Safety Valves

Among the interesting recommendations in the report of the committee on safety valves submitted at the recent convention of the Association of Master Mechanics and Master Car Builders at Atlantic City, is the following:

"Manufacturers should be required to stamp on the valve the lift in inches, as determined by actual test, with valve in working condition and set for a blow-back of not to exceed 3 lb. The committee recognizes the fundamental importance of valve lift. It therefore recommends that in design and manufacture valve lift receive proper consideration."

This opinion is interesting in its bearing on the discussion of this point in safety-valve design before the American Society of Mechanical Engineers. While advanced by those concerned primarily with safety valves for locomotives, what may be said for them applies with equal force to safety valves on stationary boilers.

Bureau of Mines Publications

The Bureau of Mines has the following publications for distribution. Only one copy of each will be sent to a person, and they should be ordered by number and title, addressing the Director of Bureau of Mines, Washington, D. C.

Bulletin No. 41: "Government Coal Purchases under Specifications," with analyses for the fiscal year 1909-10, by G. S. Pope, with a chapter on the fuel-inspection laboratory of the Bureau of Mines, by J. D. Davis.

Technical paper No. 5: "Constituents of Coal Soluble in Phenol," J. C. W. Frazer and E. J. Hoffman. No. 9: "Status of the Gas Producer and of the Internal-Combustion Engine in the Utilization of Fuels," R. H. Fernald. No. 19: "Factor of Safety in Mine Electrical Installations," H. H. Clark. No. 20: "The Slagging type of Gas Producer," with brief report of preliminary tests; C. D. Smith.

Bulletin No. 39: "The Smoke Problem at Boiler Plants," a preliminary report; D. T. Randall; a reprint of United States Geological Survey Bulletin No. 334, revised by S. B. Flagg.

J. Paul Clayton's "New Analysis of the Cylinder Performance of Reciprocating Engines" has been issued as Bulletin No. 58 by the engineering-experiment station of the University of Illinois. The presentation in the bulletin is rather more complete than in the paper submitted to the American Society of Mechanical Engineers, and has appendices, describing in detail the construction of the logarithmic diagram, the experimental plant, the methods of test followed, and a bibliography of the subject. It may be had upon application to W. F. M. Goss, Director, University of Illinois, Urbana, Ill. Price, 60c.

BOOKS RECEIVED

CENTURY LECTURES. American Gas Institute, New York. Cloth; 174 pages 6x9 in.; illustrated; indexed. Price, $2.

THE PRINCIPLES OF HEATING. By William G. Snow. David Williams Co., New York. Cloth; 224 pages, 5¾x9 in.; 60 illustrations; tables; indexed. Price, $2.

ANALYSIS OF METALLURGICAL AND ENGINEERING MATERIALS. By Henry Wysor. The Chemical Publishing Co., Easton, Penn. Cloth; 82 pages, 8½x10½ in.; indexed. Price, $2.

CHEMICAL ARITHMETIC AND CALCULATION OF FURNACE CHARGES. By Regis Chauvenet. J. B. Lippincott Co., Philadelphia, Penn. Cloth; 302 pages, 5¾x9 in.; tables; indexed. Price, $4.

NEW PUBLICATIONS

ASSOCIATION EDUCATIONAL WORK. By George B. Hodge. Published by the Association Press, New York City, 1912. Cloth, 256 pages; 5¼x8 in.; illustrated. Price, $1.

This book is the result of much painstaking effort by Mr. Hodge and represents 20 years of experience in Y. M. C. A. endeavor to organize associations, give men and boys occupational training and decrease the alarming growth of illiteracy in the United States, due to immigration.

Reliable statistics are given to show that there are over 370 different kinds of professions, occupations and trades in America, only a small portion having definite facilities to help men and boys prepare for their life work.

Strong emphasis is laid on the necessity of a closer relation between the shop and the school, and in all the day courses recommended for boys between 14 and 22 years of age half of the time must be spent in the shop. Special attention is directed toward helping a young man to choose that vocation for which he is mentally and physically fitted, and the association claims to have been very successful along this line. "The boy without vocational guidance," says the author, "is father of the bench-warmer and the unemployed."

To those interested in vocational and moral uplift this book will prove valuable in practical suggestions of field study, financial problems, short cuts to educational development and making good citizens of boys who are unable to help themselves. A complete cross-reference index of over 500 topics treated is placed at the end of the book.

A PRACTICAL MANUAL OF STEAM AND HOT WATER HEATING. By Edward R. Pierce. Domestic Engineering Co., Chicago. Cloth; 339 pages; 5½x8½ in.; 30 illustrations, tables. Price, $2.50.

The book is primarily for the practical man. The mathematical treatment is simple, involving only arithmetic. It should be appreciated by steamfitters and practical men engaged in house heating. In the steam tables the older values are used. The book would have been more uptodate with the new Marks and Davis tables.

Historically, the book is interesting for its material on the early forms of radiation, cast-iron boilers and their progenitors. Considerable space is given to the changed conditions of the house-heating business whereby the steamfitter instead of the boiler manufacturer has become responsible for the proper installation of heating systems. Mr. Pierce mentions the fact that cast-iron boiler manufacturers have raised their ratings due to competition and furnish practically no information to enable the steamfitter to select the proper size of boiler for any given condition. He recommends that engineering societies interest themselves in the establishing of a standard for information to be furnished by cast-iron boiler manufacturers in their catalogs. It is proposed that radiation be sold on a guaranteed B.t.u. basis due to the widely ranging efficiency of the different types of cast-iron heating surface. The book is commended to society members interested in standard ratings for cast-iron boilers.

The use of Wolff's rule is recommended for determining the necessary radiation instead of rule-of-thumb methods. The element of air change in this method makes an uncertainty, giving an excess of radiation for large rooms while generally correct for small spaces. The method, however, is recommended in place of an inexperienced guess.

One section is devoted to explaining how the barometrical difference in altitudes would affect the boiling point of water and possibly the boiler capacity. This should have nothing to do with the capacity or boiler output, although steam would be generated earlier and at somewhat lower temperature at higher altitudes. A pressure of 2 lb. at the boiler in the catalog should mean 2-lb. gage pressure with the barometer 29.9 in. and 218.5 deg. F. This is an absolute pressure and temperature within the system whether on Mount Shasta or New York. Some open vapor systems might have to be modified by closing the circuit entirely, but this would not affect the rating or capacity of the boiler.

A table given shows a difference of 10 deg. between the steam temperature of the boiler and the radiation. If this means the steam temperature in the radiator, there would be a drop of 3.2 lb. from 219 deg. to 208 deg. in the radiator and 4 lb. from 227 deg. in the boiler to 215 deg. in the radiator. This would require a fall of 9 ft. between the lowest radiator and the water line of the boiler to prevent flooding, and would be difficult to provide for in many house-heating systems. These low radiator temperatures may be another way of making sure of sufficient radiation. A temperature of 185 deg. is given for the boiler and 180 deg. for the radiator while the items either side show a difference of 10 or more degrees.

The discussion on continuous operation of heating systems is commended to all who may be interested in large or small plants but it applies most forcibly to the former. The author indicates the effect on the heating plant of cooling the masonry walls and the increased cost of intermittent operation as well as the increased capacity necessary.

On another page are three arrangements of heating surface at different temperatures, the highest temperature costing the least, and the author says that the higher pressure will be the most expensive to operate. In view of the statement previously made that pressure is only a measure of intensity and not of capacity, this is not so if the pressure and radiation are designed for zero weather in each case and a boiler of the same capacity is installed. One arrangement will be just as economical as the others in proportion to the ease with which the pressure and temperature of the steam can be changed to meet outside conditions of temperature. If any of the arrangements were maintained at the maximum pressure stated, the rooms would overheat in moderate weather and fuel would be wasted just as ordinarily happens where the maximum is 2 lb. pressure unless automatic heat control is provided. If the boiler was selected by the catalog rating of surface as called for at the higher pressure, it would be too small and Mr. Pierce's statement would be true.

Vapor systems, it is declared, are not well fitted to operate on indirect heating due to low temperatures and, it might be added, large steam volumes as well. As it is generally desirous to heat living rooms with fresh air supply in this class of work the higher pressures will be found more effective.

Section XIV discusses the arrangement of the main return with reference to the water line of the boiler. This is mentioned as Table FF cannot be reconciled with the conditions given.

Another section outlines the information necessary in boiler catalogs so that a proper selection of boiler may be made for any given case. Years ago it was customary to guarantee that heating plants would maintain a room temperature of 70 deg. in zero weather without attention to the fire for eight or ten hours, and nothing was said about the pressure. Table G2 gives some interesting data as to the effect on the rating if this kind of guarantee were required.

Comparing two boilers, one with large flues and small fire surface and one with the reverse, both having the same total heating surface, the author reports tests under like conditions which showed that the second transmitted almost twice as many B.t.u. in a given time. The grate area, amount of fuel and duration of test

are not stated, but would have been interesting so as to determine the relative efficiencies.

The book is written as though the author were conversing with the reader, which relieves the monotony of what otherwise might prove dry reading and maintains the interest. In addition to the useful information and simple instruction, the book contains some pointed cartoons on false boiler ratings. It is to be hoped that the book will have a decided influence in placing bolier ratings on a rational basis.

EFFICIENCY AS A BASIS FOR OPERATION AND WAGES. By Harrington Emerson. Third edition, revised. Published by the Engineering Magazine Co., New York. Cloth; 171 pages; 7½x5 in. Price, $2.

For the most part this is purely a philospohical treatment of the application of efficiency methods in their relation to the various phases of modern industry.

The first half of the book is devoted to the philosophy of efficiency; the remainder, with the exception of the last two chapters, is more specific and deals with the actual work of the line and staff organization; cost reduction, accounting and standards; waste elimination, and efficiency-system operations. Standard time and bonus systems are also well treated.

In the first chapter appears the assertion: "The actual and potential wastes in each year amount to as much as the total accumulation of wealth." This is supported by the statement that by proper management the pig-iron output of one foundry was increased 500 per cent., and that $300,000,000 per year is the cost of preventable wastes in American railway practice.

Americans are severely criticized for lack of "patience, persistence and things enduring." Mr. Emerson is not borne out by history, so far as our lack of the second mentioned quality is concerned, and if he is right in the rest, it is not to be overlooked that it is this very spirit that is responsible for our rapid, though perhaps extravagant progress, and for our international prestige. Our further success depends on our adopting corrective measures to offset the evils of prodigality.

The need of complete parallelism between line and staff is admirably brought out. Power costs, it is said, may possibly be reduced 28 per cent. and power consumption 30 to 33 per cent. This applies to many plants, but by no means to all.

As an example of time losses in labor operations it is stated that: "Standard times may be anywhere from 10 to 90 per cent. less than actual times." The importance of making it pleasant for the worker operating under standardized time is emphasized. Unfortunately the worker is more often regarded as a unit machine whose psychological attributes are an unavoidable impediment to the more rapid increase of profits. This is not the fault of the efficiency officer, but rather the management.

Chapter 10 dwells on the necessity of equity in connection with bonus systems. The inefficiencies of overtime work are well pointed out. The remaining two chapters are philosophical.

The book is well arranged and commendably carries out the author's intention to supply those seeking a knowledge of the fundamentals of "efficiency."

GOOD ENGINEERING LITERATURE; What to Read and How to Write. By Harwood Frost. The Chicago Book Co., Chicago, Ill. Cloth; 422 pages; 5¼x7¾ in. Price, $1.

The purpose of this book is fourfold: to prove the value of good English and literary expression; to show how to collect and arrange engineering subjects for the press; to so aid the engineer in reading literature that his time and money may be profitably spent, and to collect and preserve information which is unattainable to the average engineer.

While Mr. Frost's book is mainly written for the professional engineer, it can be profitably studied by operating engineers who hesitate to write for their trade and technical papers because they fear that their productions are not properly constructed or lack an "elegant flow of language."

The chapter, "What to Write About," urges the contributor to aim to make his contribution "practically useful to his brethren; its merits depend on his success in doing this." The technical paper is "always looking for information on new and better tools, devices and processes, and then makes it public for the general benefit."

Chapters are also devoted to the preparation of manuscripts for publication, grammar, punctuation, card indexing and filing, and the engineer's library.

Principe di Piedmonte Explosion

Contrary to some reports, the accident on the Lloyd Sabaudo liner "Principe di Piedmonte" while at sea on July 17, in which five men were killed, was not due to an exploded boiler tube or valve, but to a bursted supply pipe from a boiler to the header.

The five boilers are connected to the header by bends of 6-in. copper pipe. The break occurred in the copper pipe at the flange connection to the header, and between the header and the stop valve. A close examination of the break showed it to be fresh and more clean than jagged.

Fourteen men were in the boiler room at the time, nine being at one end near the engine room door. The remaining five, who were killed, were trapped between the affected pipe and the door, and as the steam rushed out of all five boilers they were immediately overcome.

No cause is assigned for the break. Nothing is said to have hit the pipe and, unless something was radically wrong, water-hammer could not have caused it.

Those killed were the first engineer, the chief fireman and three stokers. It was the chief fireman's first trip on the vessel, he having transferred to her in order to visit his son in New York.

The accident is identical to that of the "North Dakota," which burst a 7-in. pipe during an endurance test in 1909. Editorial reference is made to the accident on page 160.

The Ruppert Accident

In the haste of going to press with the report of the accident at the plant of the Jacob Ruppert brewery, New York City, in the issue of July 16, proper verification of the facts was omitted. It appears that the report as given contained certain inaccuracies.

Instead of a cylinder head being blown out of the ammonia end of one of the compressors, four bypass valves were blown out. Soon after the escape of the ammonia fumes an explosion occurred which killed one man instantly and afterward resulted in the death of two others.

The direct cause of the explosion or the bursting of the bypass valves is not known. It is thought that the ammonia mixed with the air to form an inflammable gas which was ignited by the flaming arc lamp in the engine room. Although damage to the woodwork and electrical fittings was quite serious, the plant was again put in operation soon after the ammonia fumes cleared away.

Electric Shock Fatal

Attempting to fight a fire in the tower of the City Hall, Atlantic City, on July 17, Capt. Edward Barnett, of the fire department, was instantly killed when a lantern in his hand completed a high-tension electric circuit. Chief George Profatt, of the City Electric Bureau, was knocked unconscious in an effort to rescue Barnett, and several firemen were hurled from the tower to a landing 15 ft. below when the metal trimmings of the tower on which they were standing became grounded.

Elevator in 70-ft. Drop Seriously Injures Six Men

Six men were seriously hurt when the cable of a freight elevator in the Puck Bldg., Lafayette and East Houston Sts., New York City, snapped and dropped the car from the third floor to the subcellar 70 ft. below. The safeties were evidently not in working order.

Engineer Killed by Escaping Steam

A serious boiler accident recently occurred at the plant of the Union Brewing Co., Nanaimo, B. C. Preparatory to removing the stays from the dome of one of two boilers connected together by a line containing a slip expansion joint, the bolts were removed from the flange connection of the line at the idle boiler. As the other boiler was under full pressure, the expansion joint, which evidently was not provided with studs and checknuts, let go as soon as the last bolt was removed. The escaping steam overcame and killed Albert Randle, one of the assistant engineers. Editorial reference is made to this accident on page 159.

New River Development Inspection

General Manager H. W. Fuller, of the Appalachian Power Co., Bluefield, W. Va., was host recently to over a hundred visitors on a trip to developments Nos. 2 and 4 of the company on New River. A special train left Bluefield early in the morning and returned in the evening. The party was taken over the developments and the work described.

It was announced that development No. 4 will be ready to furnish power, Aug. 1, and development No. 2, Oct. 1. The former will have a capacity of 9000 hp., and the latter 20,000 hp. A handsome booklet giving the company's history and photographs and statistics showing the progress of the work to date was given each guest as a souvenir.

N. A. S. E. Convention Program

The local committee of the National Association of Stationary Engineers, which has charge of the arrangements for the coming national convention, to be held in Kansas City, Mo., during the week of Sept. 9, has prepared an excellent program, and unless all signs fail, this year's convention will be particularly notable.

Monday, Sept. 9, will be given over to the usual receptions of delegates, members, visitors and their ladies, registration and the distribution of badges at the headquarters, the Coates House, where the local Ladies' Auxiliary will tender an informal reception to the visiting ladies. In the evening the national officers will receive the delegates and visitors in the hotel parlors; an informal musical program will be offered.

At 9:30 a.m., Tuesday, W. F. Groom, chairman of the local committee, will call the convention to order in the New Casino Hall, at 1023 Broadway. An imposing list of speakers has been prepared for this session. Governor Herbert S. Hadley, of Missouri, will give the address of welcome and National President E. H. Kearney will respond. Governor W. R. Stubbs will welcome the visitors to Missouri's sister state, Kansas, and National Vice-President J. F. McGrath will then respond. Mayor Henry L. Jost, of Kansas City, Mo., will welcome the delegates to the convention city, Past National President Joseph F. Carney replying. John H. Atwood will speak on "The West" and Past President Carl S. Pearse will respond. At high noon the convention will be officially opened by National President Kearney, who will appoint the various committees and make the formal announcements. The official photograph will then be taken at the south front of the convention hall. At 12:30, the mechanical exhibit in Convention Hall, Thirteenth and Central Sts., will be formally opened by National Vice-president McGrath. At 2 p.m., an official session will be held, and in the evening the visiting delegates and members will visit the city's fraternal organizations.

Wednesday's official session will be held at 9 a.m. At 1:30, special cars will leave the Coates Hotel for Electric Park, where a complimentary entertainment will be given the visitors during the afternoon and evening by the owners, J. J. and M. G. Heim, under the personal supervision of C. H. Mann. Admission will be given only to wearers of the official badge and their families. The annual baseball game between the engineers and exhibitors for a silver cup will be played at the Gorden Koppel stadium, near the park, at 3 p.m.

On Thursday, the official session will be held at 9 a.m., followed by the annual memorial services at 11:30. At 2 p.m. the fourth session will be held. In the evening the usual entertainment by the National Exhibitors' Association will be given in Casino Hall.

On Friday there will be two business sessions and the election of officers. In the evening the officers will be installed, followed by a grand ball in New Casino Hall.

On Saturday the annual meeting of the officers will be held at the Coates House. Visitors will be escorted to the various places of interest about the city upon request to the information committee.

During the convention the Ladies' Auxiliary will hold several sessions at the Coates House, interspersed with an attractive program of entertainment, including an automobile ride on Thursday at 2:30 p.m. Automobiles will leave the hotel at 2:30, proceeding to the Swope Park Zoo by way of Penn Valley Park and sections of Kansas City's splendid boulevard and parkway system. The return route will be along the famous Cliff Drive.

National Association of Cotton Manufacturers Meeting

The ninety-third meeting of the National Association of Cotton Manufacturers is to be held at the Griswold, Eastern Point, New London, Conn., beginning on the evening of Sept. 9, 1912, and continuing through the two following days.

The delightful scenery of this location and the attractions of the house are such that an attendance of over 400 is expected.

The program of the meeting is in an advanced state of preparation and will not be issued until shortly before the meeting. The association desires from the members further papers on subjects connected with the cotton manufacture or tributary to it.

SOCIETY NOTES

The second annual meeting of the Institute of Operating Engineers will be held on Friday and Saturday, Sept. 6 and 7 in the Engineering Societies Building, 29 West Thirty-ninth St., New York. Friday morning will be devoted to the reunion of members and friends and the reports of committees. The main features will be the award of diplomas on Friday evening and the report of the Constitution Revision Committee on Saturday. Friday afternoon will be devoted to the reading and discussion of papers.

PERSONAL

President R. C. Maclaurin, of Massachusetts Institute of Technology, has sailed for Europe to spend his vacation largely in inspecting the buildings and equipment of technical institutions at Birmingham, Manchester, Glasgow, Munich, Charlottenburg and elsewhere, preparatory to the building of the "New Technology by the Charles."

It is not unlikely that Joseph McNeil may resume his connection with the Massachusetts Board of Boiler Rules, but in a different capacity. Robert J. Dunkle, who represented the boiler-insurance interests upon the Board, is connected with the Casualty Company of America, which company has given up boiler and flywheel insurance. This will probably involve the substitution for Mr. Dunkle of the representative of a company which is actively interested in that branch, and it is entirely likely that Mr. McNeil, who resigned as chairman of the Board of Rules to accept the chief inspectorship of the Eastern district for the Hartford Steam Boiler Inspection & Insurance Co. may be the man selected.

Moments with the Ad. Editor

Some weeks ago we asked readers to tell us something of their experiences with Power advertisers.

We wanted to know *definitely* what they had bought or recommended the buying of that was installed in their plants and that was first called to their attention through Power.

We offered to pay for the letters published and here is number one, from A. R. Kummerer, of Hazleton, Pa.:

"Correct advertising is the best way for the manufacturer to put his goods before his public. For these reasons:

"First—By advertising in a paper like Power with a good live circulation, he can state the merits of his goods to thousands of people who are looking for things to save them money and who would not find them with the goods themselves stuck away on the shelves of supply houses. He can do this without the loss of time and without the inconvenience required to go the supply house. And in *one issue* of Power.

"Second—Seeing goods advertised continuously sets a fellow to thinking that there must be more than talk back of the goods or they could not stay because papers with live circulations don't carry advertising free of charge.

"Third—The illustrations of advertised articles coupled with the manufacturers' generous offers of catalogs make it convenient for the engineer to compare the articles with the conditions in his plant.

"I have purchased Cling Surface Belt Dressing, Schieren Duxbak Belting, Manzel Bros. Oil Pump, Keystone Grease, Smooth-On Iron Cement, McCord M'f'g Co., Gaskets, National Feed Water Heater, Kewanee Unions of National Tube Co. Sheet Rubber of Quaker City Rubber Co., and have an order placed with the Coppus Engineering Co., for a turbo blower of their make.

"Personally, I have purchased books on engineering work, namely, Engine Room Chemistry and Steam Turbine Practice and will continue to buy such books as are of value to me.

"I never omit looking over the advertising matter in Power and often pick up old issues and look over the ads to make sure that I have not missed something that would be of value and service to me."

Now here is *your* opportunity to tell us something about *your* experiences.

Every letter published is paid for in real money, as are all contributions to Power. Letters on this page command from $2.00 to $5.00.

For several years the ad. editor has been preaching that it pays to read advertisements and buy advertised goods.

Somewhere back in the dust another power-plant paper has come out of the tall grass and is following in father's footsteps.

It has made a very careful study of father's language, too, so that if past performance argues for future action, it will, within the next few weeks, ask *its* readers this self-same question—What are you getting out of advertising in *that* paper?

Now, since the ad. editor holds no patent-right, copyright or moral right to the English language, all he can do is to ask the question *first* and then leave it to Power readers to make a *quick* response.

If you will tell him *now* what you have gotten out of Power advertising you will be paid for every line printed.

Let us know definitely what you have bought or influenced the buying of for your plant that was first brought to your notice through Power.

Address the Ad. Editor, Power, 505 Pearl Street, New York.